JAMES WATT
The father of the Modern Steam Engine.

PUMPING ENGINES

FOR

WATER WORKS

BY

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DEDICATION

THIS BOOK IS DEDICATED TO MY FRIEND
AND PHYSICIAN

DR. JOSE F. DE F. FERNANDEZ.

PREFACE

The writer is well aware of what it means to add another book to those already in existence upon the subjects of Engineering, Mechanics, and Machinery, and therefore feels somewhat obliged to say a word or two which might seemingly justify his position in so doing.

This history of pumping machinery to a very limited extent; this setting forth of a few general principles; this description of the leading types and classes of engines used for pumping water for municipal or public supply; in short, the contents of this book are not produced for the purpose of introducing any specially new subject matter, but rather because it is desired upon the writer's part to make an attempt to attract the interest of the great mass of students, engineers, water works managers, and the men who attend to the practical operation of pumping engines in service, by presenting to them a well known subject in a form which it is hoped will hold their attention in a pleasant and profitable manner, besides encouraging them to think about what they are reading.

Books written upon subjects kindred to this one have been in many cases handled in a masterful manner from time to time, and by very high authorities. But there is a general feeling abroad, approaching, if not even quite amounting to a certainty, of aversion to deep and complex mathematical explanations and exhaustive and exhausting technical discussion, with which text books and reference books abound. Therefore it is proposed herein to exhibit the work without a display of the tools by means of which such work has been accomplished, and it is supposed by the writer that this course

will excite the greatest attention, and give the most of useful facts, with the least labor to the reader. This, accompanied by the touchstone of personal experience, will, it is hoped, make a strong appeal to human interest.

Those about to take up or to extend the study of steam engines, pumps, and pumping engines, are often discouraged at the very start from approaching the matter as closely as they desire, by the complicated character of the opening pages of a great many of the text books; and, although natural laws and their principles and effects must be necessarily wrestled with to a very serious extent by the student at school, and others later, by way of substantial foundation for after life, it seems as though there might be a small space available for a book which deals with the subject in a somewhat primitive manner, in plain every-day words, without taking too much knowledge for granted upon the part of the reader, and with the almost total absence of mathematical details. If the student or the layman hesitates at the smaller books, some of which are really too brief upon important points and often assume too much knowledge of principles and details upon the part of the reader, then it may be imagined with what thoughts and feelings the beginner or the seeker after knowledge approaches the larger books, which go so thoroughly into minute details. Besides all this, even a person of experience likes to see something in print which clearly and unmistakably coincides with his own views. It makes him feel that he is not mistaken in his own course and ideas, which is more or less assuring to anyone.

In the following pages it has been considered best to commence at once, after a few words of introductory character and of historical interest, with a description of the highest attainments of pumping engines and a comparison between the nearest practicable approach as shown by the Mariotte curve, to theoretical lines in steam expansion, and the actual practice of building and operating such machinery upon a large scale. In this way a sort of reflected object lesson is brought

before the mind with a very near approach to a practical demonstration of the principles involved; and after this introduction, bringing in only the most important points such as initial or boiler pressures, ratios of steam expansion, theoretical and best actual vacuum, receiver and counter pressures, steam jackets and the amount of steam used or wasted in such jackets; the subject is then carried along natural lines touching upon the relations between steam and coal consumption, the investment value and costs of pumping plants, taking in the various types and classes of pumping engines past and present, with mention of a few of the more prominent details of construction, kinds and qualities of materials, and finally finishing with the principles and practices involved in duty tests of pumping machinery for water works service.

Now, this demonstration by means of drawings, figures, and pictures alone, is nothing like so easy or effective as contact and experience with, or even observation of, the machinery itself; and since the subject-matter herein cannot possibly be learned from books alone, it is strongly recommended by the writer that the thoughts the reading of this book may give rise to, be supplemented by the observation of pumping engines in operation, and as much practical experience as it may be convenient or possible to acquire. In fact it is assumed that this book is to be used either to finish up a course at school or college, or to assist a practicing engineer, or a running engineer at a pumping station, or a fireman in the boiler room, or water works managers and superintendents, to understand what can be done with a pumping engine under various conditions and in different situations.

The view has been taken herein, that the reader, especially if he is a beginner, should not be confused at the start by mentioning numberless possibilities and conditions, no matter if they may be extremely interesting on their own account. The aim has been rather to present types and classes, and to impress upon the reader's and the student's attention the fact that

after all, such types and classes of machinery are subject to numerous modifications and variations in order that the ever changing and differing conditions found at different pumping stations may be effectively met to the best advantage, all things considered.

CHARLES ARTHUR HAGUE.

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CHAPTER I

THE PUMPING ENGINE

WHEN a substance, as, for example, coal, goes through the process of burning in a furnace and makes what is known as fire, a particular form of energy is produced termed heat; and such a process is called combustion.

If the heat is applied to a fluid, as, for example, water, a form of gas is produced; and such a gas is called steam. The temperature at which steam is produced varies with the pressure; at ordinary atmospheric pressure the temperature is 212° Fahrenheit.

When steam is formed within a vessel not closed, as, for example, a teakettle, the pressure will remain that of the atmosphere, and the temperature at 212° Fahrenheit. If the steam is confined in a closed vessel, as, for example, a steam boiler, the pressure can be utilized; and the greater the amount of heat applied to the boiler, the more steam will be made, the greater the amount which can be utilized, and the greater the temperature and pressure will be.

From the boiler the steam can be conveyed through a pipe to a closed receptacle known as a cylinder; and within this cylinder can be placed a disk called a piston, fitted to a rod; the piston is capable of moving freely in the cylinder. The steam can press the piston forward and backward, or upward and downward, as the case may be, within the cylinder, and so produce motion of the piston rod.

The motion of the piston rod, impelled by the piston, which in turn is driven by the pressure of the steam within the cylinder, is employed in many ways as shown by the great variety of forms and work of the steam engine. In this book, however,

the work of the steam engine is considered only with reference to the pumping of water for public supply, and in the form of steam machinery known as the water works pumping engine, which is a machine consisting of one or more steam cylinders and one or more water cylinders, with appropriate supports, framing, and working parts, so as to utilize the heat energy in the useful work of pumping.

The first duty of a pumping engine is to PUMP WATER; to do this successfully and continuously. And it matters little what else it can do if it cannot do this. If it fails in this, it fails in its most important mission and in its reasons for existence.

Next to reliability and durability in continuous operation, comes economy in steam consumption. And to combine reliability, durability, and steam economy, a great deal of thought, study, and effort, have been expended upon the pumping engine for more than a hundred years.

There have been pumping engines designed and built in which the repairs have been excessive, and the necessary stoppages have been costly, troublesome, and annoying; although when actually in operation steam economy has been reasonably high. Some engines meeting fairly well the requirements of steam economy, but lacking in durability, have something radically wrong in their makeup or in their adaptation to the conditions of the work to be done.

The necessity for staying qualities must be clearly recognized at all times, although at each successive step along the line of changed economic conditions, new, or at least different difficulties in the matter of construction and of working forces have been successively met and provided for, until to-day it looks very much as if the practical and indeed the theoretical limits of design and efficiency have been at last reached.

There have been pumping engines designed and built in which the repairs have been next to nothing, or at least at a very low rate, although such pumping engines have been in nearly continuous service for over twenty years. Such engines

have met to a very satisfactory degree the requirements of reliability and durability, but have been considered low in steam economy.

And so for many years there has been urged a sort of engineering and commercial warfare between the contending forces representing low and high economy in steam consumption, or, according to the old war cries so well known and remembered by those who were active in the water works field twenty years ago, "low duty" and "high duty."

The simplicity and directness of the best types of the low duty pumping engines have never been equaled by the high duty machines. Simplicity of construction seems to be a companion of low duty; and it seems to be impossible to separate comparative complication from high duty. The value of coal, the value of capital, and the daily capacity per unit, have been important factors in the problem. And so the contest has gone on until with greatly increased pumping capacity in a single engine; greatly modified cost in construction, partly from simpler designs and partly from better shop management; greatly reduced rates of bond interest ranging from 8 per cent in 1873, down to 3½ per cent at the present time; station duties based on all coal burned, ranging from 111,000,000 to 135,000,000 annually, with a low rate of repairs; with all these and some minor items, the high duty pumping engine has forged ahead and to the front, and the low duty engine has been relegated to rather small capacities where fuel is very cheap.

But of course Rome was not built in a day, and no matter how clearly the higher attainments may now appear, the road to these attainments in modern pumping engineering has been long, expensive, and troublesome to travel, both for builders and for buyers. And as in all engineering endeavors to improve the lot of mankind by means of the adaptation of natural forces and resources to his use, notwithstanding the hard work, occasional fallings out, and even threatened warfare so to speak, the inherent and everlasting good will which is really at the bottom of industrial progress persistently holds its own.

It is not the purpose of this book to consider what at the present time would be termed ancient methods of raising water, although the fact was evidently recognized a long time ago, that progress in the well-being of mankind depended a very great deal indeed upon a plentiful supply of water; good and wholesome if possible, but water at all events. The particular point of good and wholesome water did not then have to be driven home so incisively for the good and simple reason that the opportunities for pollution and unwholesomeness were much less frequent centuries ago than now, at this time we call the present, when the great concentration of inhabitants now to be observed as taking place in all parts of the civilized world.

No attempt has been made herein, in fact quite the contrary, to exhibit the tools so to speak, such as intricate rules and formulæ, with which the work of the book has been accomplished; and all statements of facts, and all views of the writer, have been brought down to plain and simple language so far as the technical nature of the subject will permit. Further, it is not the purpose, and it is not hoped, to teach very much to the regular designers and manufacturers of pumps and pumping machinery, although judging from past performances, a hint or two in their direction now and then would not be amiss; and, therefore, all ideas of giving instruction to designers and manufacturers will be limited to the pointing out of the fact that a dead engine in the shop, being built and put together in a sort of preliminary way, and necessarily so of course, is a totally different animal so to speak, from the live machine under steam and water pressure, doing work in its actual field of usefulness. This difference does exist, as the writer knows from experience during something like thirty years, and embraces all kinds and qualities of designers and builders. None escape. And it is useless and a waste of time to try to escape entirely. The thing to do is to carefully design a pumping engine to suit the conditions presented, and see that the designer gets the right conditions; have his drawings made by experienced men, have the work put through the foundry, the

forge, the machine shop, and the different highways and byways of construction, ship it to its destination and erect it upon the foundations prepared. And, whatever in the way of human weakness there may be reflected in the machine, put it down in the experience book for future avoidance or use, do the best that can be done and let it go at that. All seem to fare alike, the commercial product from the highest sources, the special design specially built at fifty per cent advance over the commercial article, and the veriest runt of a machine that ever attempted to pump water. And this is not written in a spirit of blame; but to state a condition of things inseparable from human nature, and to let the buyer of pumping engines know the hopelessness of seeking perfection according to his pet ideas.

But there are a large number of people aside from manufacturers greatly interested in the subject of water works pumping engines. Those who buy such machinery either for their own investments or as representatives of municipal corporations; those who endeavor to advise and counsel in the adaptation of different classes and types of pumping machinery to various situations; and those who have the care and responsibility of such machinery in its daily operation. In other words this book is intended mostly for the owners of water works plants, such as water works companies; for water commissioners, boards of public works, and similar municipal authorities; for city engineers, consulting engineers, and professional engineers generally who are interested in the subject of the installation of water works engines; and for the men in the office, in the engine room, and in the boiler room, who are responsible for the economy and efficiency of the plant all along the line from the coal pile in the bunker, to the water supply in the reservoir or distribution.

The importance of public water supply can scarcely be overestimated in connection with the life and growth of cities and towns, and indeed is only limited by the value of the cities and towns themselves. Just at the present time the subject is of particularly widespread interest, covering the items of quan-

tity, quality, supply, economy, and distribution. Those fortunate communities so situated that impounded gravity supplies of water are practicable and available, are comparatively few in number; and even at that, after the main supply has been delivered for distribution, it is found that the necessity often arises for providing methods for supplying the higher districts of a city by means of supplementary or "high service" pumping. Indeed the very fact that a city is situated where hills are high enough and near enough to provide a gravity supply of water, indicates by the general formation of the country that high service conditions are extremely possible in the city itself. But apart and aside from the few cases of gravity supply, very many, in fact a great majority of cities and towns depend upon pumping for both their main and auxiliary supplies of water, and in passing it might be well to mention that the relative desirability of gravity and pumping supplies in their order of value seems to be as follows:

Gravity supply from lakes or impounded streams at proper elevations.

Gravity main supply with auxiliary pumping for high service. Reservoirs.

Gravity supply with auxiliary pumping supply for all services. Reservoirs.

Pumping from streams or lakes to storage and distributing reservoirs.

Pumping from streams or lakes to standpipes for distribution. No storage.

Pumping from streams or lakes directly to distribution. No standpipes.

The importance then of the water works pumping plant, and of the pumping engine itself may be easily comprehended by the most casual investigator of the subject. And following this line of thought the questions of desirability, capacity, type best adapted, highest practicable measure of real economy

in operation, and other appropriate details come immediately to the front. But before going into particulars as minutely and closely fitting as may be, in the absence of specified cases to deal with, it will be of interest no doubt and also a fair guide to the mind to briefly review the steps heretofore taken in the past, by the investigators and producers of pumping machinery, constantly spurred on by competition in business and by the pride of accomplishment, until to-day a point has been reached not even dreamed of three or four decades ago. And as we pause and glance backward along the roadway of progress, there rough and rocky, here smooth and pleasant, up hill and down dale, dead level and what not, it all seems very interesting and mostly satisfactory, to review and observe the testimony and evidence of the struggle which we may now safely call successful, and again pause and endeavor to realize the hard work, the gropings in the dark, and the toiling years that have gone before.

CHAPTER II

HISTORICAL

History repeats itself. The earliest written history in which heat takes steam for a vehicle in its performance of work by means of mechanism made by human beings, sets forth the steam turbine driven by the reaction of steam jets. This was about 130 years before the Christian Era. The steam turbine is again with us. But whether this portion of the cycle of events is at or near the top, or at or near the bottom, is not yet evident. The first full fledged steam turbine driving a water turbine so as to form a turbine pumping engine in the real sense of the term, and of a capacity and importance sufficient to establish its status in the water works field, is yet to come. The steam turbine and the water turbine have their proper fields of usefulness no doubt, but they have not been clearly defined.

There are in this country and in Europe eminent and highly successful engineers who predict that the steam turbine will go into hiding again within ten years. They are no doubt sincere in their convictions, but they may be mistaken or prejudiced against the turbine, although the even balance in which they have heretofore held the scales of investigation place them high up in their profession, and they have no special interest in giving an adverse judgment.

Fig. 1 shows a picture of Hero's steam turbine which as will be readily seen was of the reactionary class, and evidently would not rank very high in the scale of economy.

In 1601 the raising of water by steam power, by condensing the steam within a closed vessel was known and the fact published. The idea is that the vessel being connected by

means of a suitable pipe with a body of water, the admission of steam into an air tight vessel would drive out the atmospheric air, and then upon the condensation of the steam, the supply being shut off, a vacuum or a partial vacuum being produced, the water would ascend the pipe and fill the vessel.

In 1615 the idea was known of sending a jet of water to a great height by means of steam pressure acting directly upon the surface of the water, within a closed vessel provided with an outlet pipe for the ejection of the water.

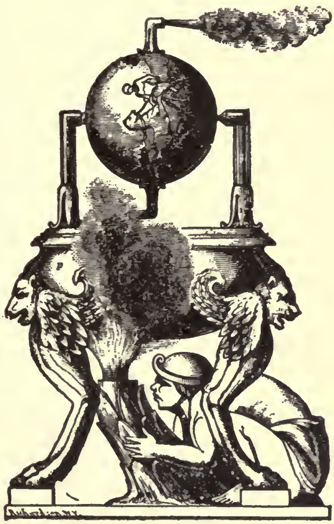


Fig. 1. — Hero's Steam Engine.

In 1629 it was well known that the impulse of steam against vanes attached to a wheel, would develop motion and power, which is of course further evidence of the steam turbine idea 1759 years after the Hero device.

In 1656, or as some records have it, 1663, there was described a machine for raising water by steam pressure, and so far as the record can be interpreted this is the first attempt to raise water by steam pressure when the steam was generated in and supplied from a vessel separate from the vessel in which the water was raised. This marks the birth of the pumping plant microbe so to speak, in which a separate "boiler" was employed to generate the steam, and separate vessels or chambers used to manipulate the water to be raised. Thus the water works pumping plant as now used in principle, has its age revealed; taking the earlier date, this is 250 years.

The Savery pumping engine with a date of 1697 seems to be the first one in which it was attempted to "lift" water by what we now call "suction" and in the same machine also lift or "force" the water to a considerable height above the engine.

So that the pumping engine as a "lifting" and a "forcing" machine combined, is only about 200 years old, or 209 according to the recorded dates. The Savery pumping engine gave a practical impetus to the hydraulic idea of raising water under pressure, and was very usefully employed for mine pumping upon what at that time must have been a liberal scale.

Fig. 2 shows a sectional picture of the Savery pumping engine, which gives a very clear idea of this early machine.

In the pumping machinery up to and including the Savery

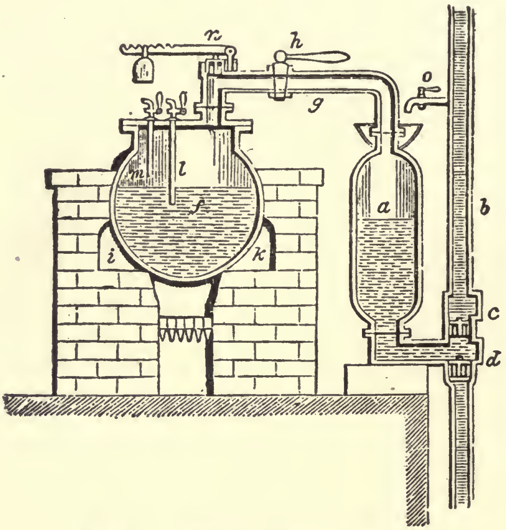


Fig. 2. — Savery's Pumping Engine.

engine, the steam acted mostly by its pressure directly upon the surface of the water and without the intervention of pistons and other communicating mechanism, as shown in the most advanced development of the Savery engine in 1697 indicated in Fig. 2.

But in 1690 the idea of the piston, now so well known, to communicate the power and motion derived from the steam, to mechanism, had already been born, and the record gives Denis Papin credit for the invention. Papin was also the

inventor of the safety valve for steam boilers, now considered absolutely necessary wherever steam is used. The piston in the steam cylinder was first exhibited in a model, and in this model the water, a small quantity, was placed in the bottom of a vertical cylinder, the piston resting upon the water. The application of heat beneath the cylinder generated steam and drove the piston to the top, whence by the condensation of the steam and the formation of a vacuum thereby, the piston was driven down again to the bottom of the cylinder by the pressure of the atmosphere, or the air we breathe; the air as many know exerts a pressure of nearly fifteen pounds per square inch against the sides of any air tight vessel containing a vacuum, or entire absence of air or vapor. From the idea of the cylinder and piston there followed the further ideas of driving pumps and rotative machinery.

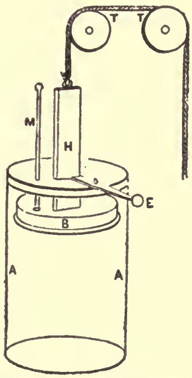


Fig. 3. — Model of Papin's Piston Engine.

Cawley, Newcomen, Savery, and others used the piston, the separate boiler, and surface condenser, and so designed the well known mine pumping engine operated by atmospheric pressure as in the Papin model. To aid the condensation of the steam, and make the engines work more rapidly and promptly, the initial form of the jet condenser was introduced, by injecting cold water into the steam cylinder itself.

Devices for making the engine open and close its own valves were soon introduced, the first by Humphrey Potter; and then the valves and valve gear were improved by Brighton.

Then Leupold came along in 1725 or thereabouts, with higher steam pressures, so improvement followed improvement, by Smeaton and others, until by 1770 this form of pumping engine in which steam was applied beneath the piston, and the atmosphere applied on top of the piston, first one and then the other, became fully up to date for its time, the state of the machinist's art, tools, and appliances generally, considered.

The following figures show pictures of the various steam pumping engines with their dates of invention or existence:

Fig. 2. Savery's pumping engine, 1697.

Fig. 3. Model of Papin's piston engine, 1690.

Fig. 4. Newcomen's pumping engine, 1705.

Fig. 5. Leupold's pumping engine, 1725.

Up to about 1770, Smeaton had perfected so far as possible what is known as the atmospheric engine, considering the uncertain and unknown lines then being followed. By the atmospheric engine is meant the engine at that time in the field, in

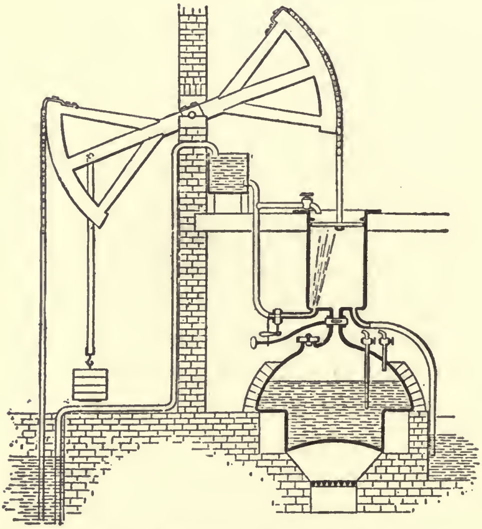


Fig. 4. — Newcomen's Pumping Engine.

which the pressure of the atmosphere on the upper surface of the piston of an open top vertical cylinder, was an important element.

In 1736 James Watt was born, and in 1759 he entered the field of steam engineering. Up to the advent of Watt the progress in steam engineering had been confined to making small alterations in specimens then in existence, and with very limited results. The steam engine as then designed and constructed represented

rude accomplishments, a very large amount of waste, and a very small measure of efficiency.

When Watt came in contact with the subject of steam, he endeavored to grasp it broadly, and according to the scientific lights of the times. In fact Watt was the first scientific steam engineer, and made a very great success of the matter. He struck out on broad principles, instead of narrowly following

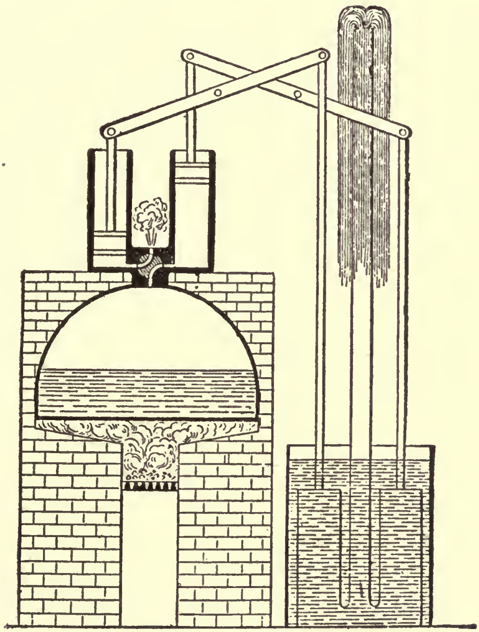


Fig. 5. — Leupold's Pumping Engine.

precedents and the then existing examples. He dealt with principles instead of details, and the details soon suggested themselves. He determined from what he could learn from the nature and action of steam, that the steam cylinder in which the steam did the work, must be kept HOT; and that a condenser, which he had introduced as a complete and separate vessel from the steam cylinder, must be kept COOL. And in carrying out these underlying principles he introduced a steam jacket, and

the separate condenser, which are used to this day. With Watt also came the cutting off of the steam within the cylinder before the stroke was finished, early or late according to conditions, and the completion of the piston's stroke by the expansion of the steam within the cylinder independently of the boilers. Also the double action engine with its closed cylinder top and a stuffing box for the piston rod to work through, and with steam driving the piston throughout both strokes.

The use of the crank which we now know so well in all sorts of machines aside from the steam engine itself, was introduced about this time to produce rotative motion of a shaft from the reciprocating motion of a piston, and its invention was disputed between Watt and Pickard, the latter however obtaining sufficient advantage in the controversy to obtain a patent on the crank, thus compelling Watt to use the device known as the "Sun and Planet" motion until the patent on the crank had expired. The crank was the better, in fact the best of all devices for the purpose, and was adopted and used by all steam engine builders where rotary motion was desired. Watt had made most of his inventions of the above mentioned details, together with some minor ones, by 1784, including the parallel motion for conveying the straight piston rod motion to the vibrating end of the working beam; the governor for regulating speed; and the steam engine indicator.

In 1769 Watt wrote a specification to cover his inventions and researches, and the writer considers them of sufficient historical interest to give them here in full, aside from the fact that to this day this specification embodies the principles which the scientific development of the steam engine are based upon. Watt's specification, with the language slightly changed in places to suit the present time, is as follows:

First: That vessel in which the powers of steam are to be employed to work the engine, which is called the cylinder in common engines, and which I call the steam vessel, must during the whole time the engine is at work, be kept as hot as the steam which enters it; first by enclosing it in a case of wood, or any

other materials which transmit heat slowly; secondly, by surrounding it with steam or other heated bodies; and, thirdly, by suffering neither water nor any other substance colder than the steam to enter or touch it during that time.

Secondly: In engines that are to be worked wholly or partially by condensation of steam, the steam is to be condensed in vessels distinct from the steam vessels or cylinders, although occasionally communicating with them; these vessels I call condensers; and, whilst the engines are working, these condensers ought at least to be kept as cold as the air in the neighborhood of the engines, by the application of water or other cold bodies.

Thirdly: Whatever air or other elastic vapor is not condensed by the cold of the condenser, and may impede the working of the engine, is to be drawn out of the steam vessels or condensers by means of pumps, wrought by the engines themselves or otherwise.

Fourthly: I intend in many cases to employ the expansive force of steam to press on the pistons, or whatever may be used instead of them, in the manner in which the pressure of the atmosphere is now employed in common engines. In cases where cold water cannot be had in plenty, the engines may be wrought by this force of steam only, by discharging the steam into the air after it has done its office.

Fifthly: * * * * * * Rotary engine.

Sixthly: I intend in some cases to apply a degree of cold not capable of reducing the steam to water, but of contracting it considerably so that the engine shall be worked by the alternate expansion and contraction of the steam.

Lastly: Instead of using water to render the pistons and other parts of the engine air and steam tight, I employ oils, wax, resinous bodies, fat of animals, quick silver and other metals in their fluid state.

Contemporary with James Watt, there lived Jonathan Hornblower, who in 1781 invented and patented the double cylinder or compound engine with two cylinders of different diameters and in some cases with different lengths of stroke. Steam

was admitted as at the present time into the smaller cylinder, and after doing work there was exhausted over into the larger cylinder and was again employed against the second piston. Hornblower soon found, especially in those days of comparatively low steam pressure, that the use of a separate condenser was necessary to the success of his engine, and this threw him into antagonism with Watt, practically placing a fatal obstacle in his path.

In 1804 Woolf again brought forward the compound steam engine, and although it was essentially the same in principle as Hornblower's invention, it often bears the name of Woolf to the exclusion of that of its real inventor. Woolf employed what was then considered high steam pressure and cut off in the smaller cylinder, beginning the expansion in that cylinder and continuing it throughout the full stroke of the larger cylinder, a plan of operation extensively used with pumping engines during the past twenty years. The application of Watt's condenser to the so-called Woolf compound engine, marked a step forward which has lasted even into the present day, and is regarded as the only material improvement of importance since the improvements of Watt. The economy of the compound engine was clear enough at the time of its introduction and up to 1814 it came along rapidly, but the steam pressures of that day were too low to enable it to compete with the single cylinder Cornish engine carrying an equally high pressure, and cutting off at an early point in the stroke, which the Cornish engine of Trevithick was enabled to accomplish by virtue of the massive pump rods then used in connection with the deep mine pumps. What is undoubtedly the chief advantage of compounding, or the use of multiple cylinders, was not thought of for many years after the invention of this form of the steam engine, and in fact was probably hidden on account of the limited steam pressures employed. That is the limiting of the range of temperature in any one cylinder.

In 1845 the compound idea was again brought to the front by McNaught, for the purpose of gaining power where it was not

convenient to put in a newer and larger engine. A higher steam pressure was used for the reason that it was no doubt seen that to gain anything with the additional cylinder this smaller cylinder must have available for its piston, a steam pressure somewhat above that formerly employed in the old and larger cylinder exhausting into the condenser. The additional power was

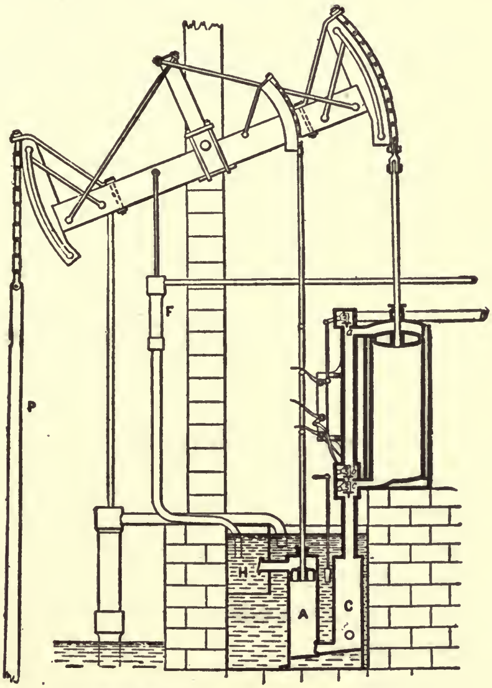


Fig. 6. — Watt's Engine, 1769.

obtained from the new high pressure cylinder, and the steam rejected by it was utilized by the low pressure cylinder, leaving the latter about under the conditions as formerly. After this original McNaught experiment was found to be successful, other engines were improved in the same way, and it was found that not only was power gained due to the actual additions to pistons and pressures, but a distinct gain in economy of steam and fuel resulted as well.

In 1850 the compound engine appeared as a water works pumping engine at the Lambeth and other water works plants. In 1854 it went into marine practice; and in 1857 the receiver between the high and low pressure cylinders was introduced with reheating facilities, thus departing from the Hornblower-Woolf alternating pistons, and opening the way for pumping and other engines, with cranks at 90° and at 120° as we find in the latest modern practice.

So it will be seen that 124 years ago most of the elements now employed in pumping engines of the highest class were known and used, and are as follows:

Steam working on both sides of the piston.
Higher and higher steam pressures.
Non-rotative pumping engines.
Crank and fly wheel pumping engines.
Steam jackets.
Cut off and expansion of steam within the cylinder.
Compound engines.
Surface condensers.
Separate jet condensers with air pump.
Poppet steam valves.
Ball governors.
Throttle valve.
Steam engine indicator.
Parallel motion for crossheads.
Crosshead and guides.
Revolution counter.

Three of Watt's engines, exhibiting the advance in his work are shown in the following pictures:

Fig. 6. Watt's engine of 1769.
Fig. 7. Watt's engine of 1781.
Fig. 8. Watt's engine of 1784.

A deeper scientific knowledge of heat and steam, together with better tools and improved materials, have combined to refine the steam engine and the water works pump, greatly increasing their economical efficiency; but after all James Watt

turned out a complete machine for the purpose in view as the above list of details employed even at the present time will testify. And to show how fully his qualities were recognized and appreciated we have only to note that a monument was placed in that most exclusive of precincts, Westminster Abbey, inscribed as follows:

Not to perpetuate a Name,
Which must endure while the peaceful arts flourish, but to show
that mankind have learnt to honor those
who best deserve their gratitude,
THE KING,
his ministers, and many of the nobles and commoners of the realm,
raised this monument to
JAMES WATT,
Who, directing the force of an original genius,
early exercised in philosophic research,
to the improvement of
THE STEAM ENGINE,
enlarged the resources of his country, increased the
power of man, and rose to an eminent place
among the most illustrious followers
of science and the real benefactors of the world.
Born at Greenock, MDCCXXXVI.
Died at Heathfield, in Staffordshire, MDCCCXIX.

There was very little indeed in the line of theory to influence or guide the early inventors of the steam engine. Watt had some advantages through his associations at the Glasgow University, with reference to the doctrine of latent heat, but the relation of work to heat was not developed into a philosophical proposition until a considerable time after Watt's inventions and determinations. It was after Watt's death that Carnot in 1824 published the theory of the steam engine as a heat engine, and demonstrated that heat does work only by being let down

from a higher to a lower level, but even he had not grasped the full idea of the relation of heat to work, and it remained for Joule in 1843 to demonstrate the conservation or convertibility of energy by showing conclusively that heat and work were interconvertible one into the other, he then setting a standard that the equivalent of one unit of heat in mechanical energy is 772 pounds

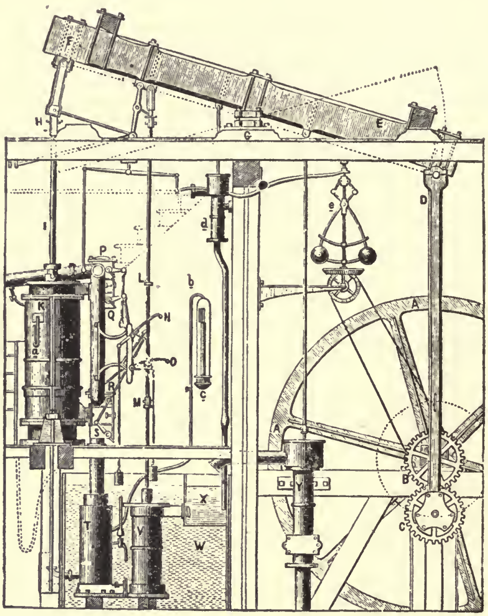


Fig. 7.—Watt's Engine, 1781.

raised or lifted to a height of one foot, a standard which is still recognized; and one unit of heat is that amount of heat which will raise one pound of water one degree by the Fahrenheit thermometer, or the thermometer we are used to seeing every day, from temperature 39 to temperature 40, at which temperature water is supposed to attain its greatest density. From 1849 the science of heat energy made great strides and was set forth by the most profound philosophers and mathematicians of our times.

The theory of heat engines taken from competent authorities may be briefly stated as follows:

A heat engine acts by taking in heat, converting a part of the heat received into mechanical energy, which appears as the work done by the engine, and rejecting the remainder, still in the form of heat. The theory of heat engines comprises the study of the amount of work done, in its relation to the heat applied and to

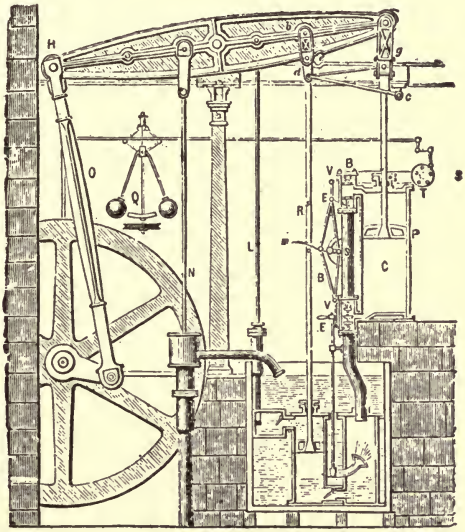


Fig. 8. — Watt's Engine, 1784.

the heat rejected. The theory is based on the two laws of thermodynamics or heat energy, which may be stated as follows:

Law I. When mechanical energy is produced from heat, one thermal unit of heat goes out of existence for every 772 foot pounds of work done; and, conversely, when heat is produced by the expenditure of mechanical energy, one thermal unit of heat comes into existence for every 772 foot pounds of work spent.

Law II. It is impossible for a self-acting machine, unaided by any external energy, to convey heat from one body to another at a higher temperature.

CHAPTER III

ECONOMIC STEAM DUTY

The highest economic duty in proportion to first cost and maintenance determines the most desirable pumping engine. And ever since the early days of Newcomen and Watt there has been a continual advance in design, construction, and performance as expressed in economic "duty" of pumping engines, accompanied by a reasonable measure of durability in the machine, as the physical conditions imposed became more and more understood. The increase in the number of opportunities for usefully employing the energy of heat, and the decrease in the chances for wastefulness from purely mechanical defects, have contributed very effectually towards the improvement from 5,000,000 foot pounds duty in the old time Savery engines, upwards to the 12,000,000 duty mark of Smeaton and Newcomen; then to the 20,000,000 duty of Watt's engines, then to the 30,000,000, then to 60,000,000 duty, as the later achievements of the Cornish engine came into the record. And so on up to 80,000,000 duty, until finally in the latter half of the nineteenth century, after say 1860, the higher steam pressures and the higher ratios of steam expansion carried the duties of the pumping engine up to past the 100,000,000 the 110,000,000 and the 120,000,000 duty marks successively; finally reaching very nearly to the possible limits to-day, a little over 181,000,000 foot pounds of work per 1,000 pounds weight of dry saturated steam consumed; that is, steam theoretically perfect, containing the exact amount of water and heat which carefully made experiments have determined to be correct according to natural laws.

The record at present, as exhibited by the pumping engines produced in this country at least, and covering the practice of a

goodly number of designers, in fact the principle engineers in this line both for special designs and for what might be termed commercial machines, is given in the following table:

Present Day Duty Record per 1,000 pounds of Dry Saturated Steam.

Builder or Designer.	Duty in Foot Pounds.
Allis-Chalmers Company	181,068,605
Edw. P. Allis Company	179,454,250
Holly Manufacturing Company	173,620,000
Snow Steam Pump Company	167,800,000
E. D. Leavitt	157,843,000
Lake Erie Engineering Works	152,000,000
Nordberg Manufacturing Company	149,500,000

The following is the pumping record for duty with superheated steam as taken from some tests at Chicago, of Worthington vertical triple expansion engines of 20,000,000 U. S. gallons daily capacity each:

Average superheat. 100°	Average steam pressure. 144 lbs. gauge.	Duty. 161,718,936 ft. lbs.
Greatest superheat. 154°	Greatest steam pressure. 147 lbs. gauge.	Duty. 174,735,801 ft. lbs.

At the close of the year 1905, the pumping engine holding the world's record for general economy, was at Boston, Mass., U.S.A., in the high service station of that city at Chestnut Hill pumping station. The data of this engine are as follows:

Designers, E. & I. H. Reynolds.

Builders, Edward P. Allis Company.

Date of completion, 1897.

Location, Boston High Service.

Capacity per 24 hours, U. S. standard, 30,000,000 gallons.

Total head against plungers, 140 feet.

Steam pressure per gauge, 185 lbs.

Duty per 1,000 lbs. dry saturated steam, 178,497,000 ft. lbs.
Duty per 1,000,000 British thermal units, 163,925,000 ft. lbs.
Steam per indicated horse power hour, 10.335 lbs.
British thermal units per indicated horse power minute, 187.8.
Thermal efficiency from absolute zero, 21.63 per cent.

The indicated horse power of this engine is 802 under the above conditions, and the pump horse power, 748, giving a mechanical efficiency of about 93.3 per cent. The steam was supplied by one large boiler of special design of the locomotive type, and was fitted with economizers in the back flue built for the purpose. The actual evaporation in the boiler under working conditions and pressure when using the economizer, per pound of dry coal was 10.472 lbs. and the efficiency of the boiler when using the economizers was 86 per cent.

Duty of engine per 100 lbs. of coal, 173,869,000 ft. lbs.
Coal per indicated horse power hour, 1.062 lbs.

The average duty per 1,000 lbs. of dry saturated steam, which is steam without either moisture or superheat, as given by 26 of the highest type and class of American pumping engines up to January 1, 1906, is 162,900,000 foot pounds. The builders and designers include the following well known names:

Edward P. Allis Company.
Henry R. Worthington.
E. D. Leavitt (designer only).
Holly Manufacturing Company.
Lake Erie Engineering Works.
Snow Steam Pump Works.
Nordberg Manufacturing Company.
Allis-Chalmers Company.

The principal improvement in the line of economy in the use of heat energy as furnished by steam in pumping engines, has been brought about by a better understanding of heat and the heat engine, of the treatment and manipulation of the working steam within the engine cylinders, and by the steady increase in the steam pressures employed. This increase in steam pres-

sures may be quickly comprehended at a glance by observing the following table:

Steam Pressure in	Pounds per Gauge.
1800	5
1830	20
1850	50
1875	75
1900	125
1906	175

Again, as the pressure of steam gradually increased, its logical accompaniment, the increase in the ratio or rate of expansion took its proper place in the march of improvement. The users of steam began by taking it within the cylinders of the early engines during the entire stroke of the piston, in those first days of the steam engine, after Papin had suggested and demonstrated the idea and possibility of the piston. Then four expansions were employed in the Watt, and also in the Trevithick Cornish engines used for pumping out mines, from 1814 to 1824, the records even giving so high a pressure as 120 lbs. per square inch. But such a pressure in the light of what must have been the limited scope of steam boilers at that time, is at least questionable; and, besides it is said that Woolf's compound engine was kept out of business on account of the greater simplicity of the single cylinder Trevithick engine at the Cornwall mines, and such a statement is at least improbable if 120 lbs. pressure was employed, for the reason that whether the lessening of the range of temperature brought about by the use of two cylinders was understood or not, the inevitable and natural effect of reducing the coal consumption for the work done would have been plainly visible even though the relations between heat, steam, and energy, were still unknown.

As later perfections in engines came into existence, the rate of expansion gradually increased with the steam pressure, and went successively to 6, 8, 10, 12, 16, 20, and upwards to the pres-

ent day limit. About 5 lbs. absolute pressure, or, 5 lbs. above a perfect vacuum seems to be the low limit of terminal pressure of profitable expansion and practical conditions; and with sufficient protection against condensation within the cylinder, it is doubtful if a lower pressure than this can be obtained in the absence of leakage. Indeed it is extremely difficult to get the expansion in proper form down to so low a point as 5 lbs. above perfect vacuum. It is very difficult if not impossible to provide latent heat enough and to supply it quickly enough within the cylinder, to obtain perfectly dry initial steam or to maintain perfect evaporation without incurring losses in other directions; but the slower the strokes of the engine, the better the chances of ideal steam conditions; and this probably explains the high efficiency of the slow moving pumping engines, the type of steam machinery holding the record for low steam consumption and high heat efficiency. Perhaps the rate of speed could be brought down until there would be time enough in each stroke to begin the superheating of the low pressure exhaust, and if this did not raise the capital account too high by unduly enlarging the engine, the complete dryness of the low pressure exhaust just to the point of saturation would likely mark the most profitable rate of motion in pumping engines, especially so when it is considered that the highest mechanical efficiency of machines lies in this direction.

But the great effect to be obtained within a steam cylinder is perfect dryness of the initial steam. If this can be accomplished the rest of the stroke will take care of itself, and if it was or ever is found that initial dryness will result in superheated exhaust especially in the low pressure cylinder, the quality of the steam must be controlled so that the best practical point can be reached for all around results. Dry initial steam in the low pressure cylinder has not yet been reached, the nearest approach being something like 9 per cent moisture remaining, but the economic duty is at present so high that the small margin yet to be attained will not stand very much expense for its acquisition.

Upon the basis of 5 lbs. absolute terminal pressure, that is, 5 lbs. above perfect vacuum, the possible ideal ratios of expansion in the absence of the usual disturbing conditions would be as follows with various steam pressures per gauge:

Steam Pressure per Gauge.	Expansions.	Steam Pressures per Gauge.	Expansions.
5	4	75	18
10	5	100	25
25	8	125	28
40	11	150	33
50	13	175	38
60	15	200	43

Thus will be seen to some extent the fallacy in seeking greater economy in more rapid motion of the piston and rotative parts. Aside from the lessening of the mechanical efficiency of the machine, by disturbances in the main pumps and increased friction in other directions, the cooling due to the expansion of the steam, and the ranges of temperature within the cylinders, all due to the expanding steam itself, and the very slight comparative change in the time for such operations, in a fast or a slow moving engine, as usually understood, go for nothing. As will hereafter be pointed out in connection with the adaption of pumping engines to their surrounding, the singled out factors of high steam pressure, high rotative speed, high piston speed, and similar factors, taken by themselves in the absence of the really controlling conditions, fail to advance the efficiency of pumping engines.

The original Cornish pumping engine was a beam engine, with the steam piston connected to one end of the beam and a rather massive pump rod at the other end of the beam. The cut off of the steam took place at an early point of the stroke, and then the stroke was finished by the expanding steam and the momentum of the heavy moving parts. This was for mine pumping and counterbalancing was sometimes resorted to when the pump rods overweighted the beam at one end. When this engine was introduced into municipal water supply pumping, in the absence

of long heavy pump rods in engine house work, heavy weights were added to make up the opportunity for storing energy which was originally furnished by the long pump rods necessary in mine pumping.

The "Bull" Cornish pumping engine, the advance agent so to speak of the modern "direct acting" pump, was a modification of the original Cornish engine, and in this machine the beam was omitted, aside from a balancing beam when the rod weights were too heavy for satisfactory operation.

The economic duty of the Cornish engine has been given in times past, ranging from 30,000,000 in 1820 through the succeeding years up to 114,000,000 in 1850, per 100 lbs. of coal, although nothing particular is said about the efficiency of the boilers, and of course when duty is given in pounds of coal burned, the efficiency of the grates, the chimney, the quality of the coal, and the generating or steam making power of the boilers, become involved in the question. It must be obvious that the engine cannot possibly be responsible for the cost of the making of the steam, although the "coal duty" of an engine is often talked about, this term really meaning the "plant duty." The engine has no grates and burns no coal, but the engine is responsible for the amount of steam used in doing any certain amount of work, this steam furnished by the boiler, which does have grates, and upon which and for the use of the boiler, the coal is burned. This consideration has finally brought about the expression of the duty of a pumping engine per 1,000 lbs. of steam, and also per 1,000,000 heat units; the rate of a thousand pounds of steam, and the million heat units coming naturally enough from the idea of a hundred pounds of coal, giving an evaporation in the boilers of 10 lbs. of steam for each pound of coal burned on the boiler grates. That is to say, 10 lbs. of water evaporated into steam per pound of coal burned on the grates, is equal to 100 lbs. of coal evaporating 1,000 lbs. of water into steam. From a scientific point of view, and for the guidance of the engine designer and builder, the expression of duty per 1,000,000 heat units usually called British thermal units may

be the better and no doubt is, but for the buyer of the engine, and the buyer of the coal, the expression in terms of 1,000 lbs. of steam is in the opinion of the writer the better; and for the reason that the economies of the boiler room, the looking after all possible raising of the temperature of the feed water, the most advantageous mode of firing the boilers, and everything in fact but the actual using of the steam by the engine, naturally separates from the operation of the machinery; and the consumption of the steam whether it is made from ice cold water or from very hot water, is an extremely satisfactory way to measure the performance of the steam cylinders. The idea is to make the steam to the best advantage possible and then get the most work out of it after it is delivered to the engine.

Up to 1848, although the Watt crank and fly wheel engine had been in existence a good many years, it does not seem to have been utilized as a pumping engine to any very great extent; but in that year there was brought out in London, England, the Simpson compound beam pumping engine, with the bucket-plunger pump, and for 25 years was considered the standard pumping engine for considerable quantities of water, say, from 5,000,000 gallons daily upwards. Fig. 9 shows a picture of the Simpson engine as then constructed for the London water works, and afterwards in 1872 placed in the water works of Philadelphia, Pa., and also of Lowell, Mass., a few years later. This type of pumping engine was also repeated at Chicago, Milwaukee, Detroit, St. Louis, and other American cities, and was generally credited with a duty of from 75,000,000 to 90,000,000 foot pounds per 100 pounds of coal consumed in ordinary good boiler practice. The West Side pumping station of Chicago, at the junction of Blue Island Avenue and Twenty-second Street, has the largest examples of the Simpson pumping engines in this country; and in their day, in the early eighties, they were no doubt the most notable pumping engines this side of the Atlantic. Their dimensions are as follows:

High pressure cylinder, 48 inches diameter.

Low pressure cylinder, 76 inches diameter.

High pressure stroke, 72 inches.
Low pressure stroke, 120 inches.
Diameter of pump bucket, 51 inches.
Diameter of pump plunger, 36 inches.
Stroke of bucket and plunger, 120 inches.

These engines continued in service a good many years and gave very satisfactory general results, the duty usually developed by them being in the neighborhood of 85,000,000 foot pounds

3491.

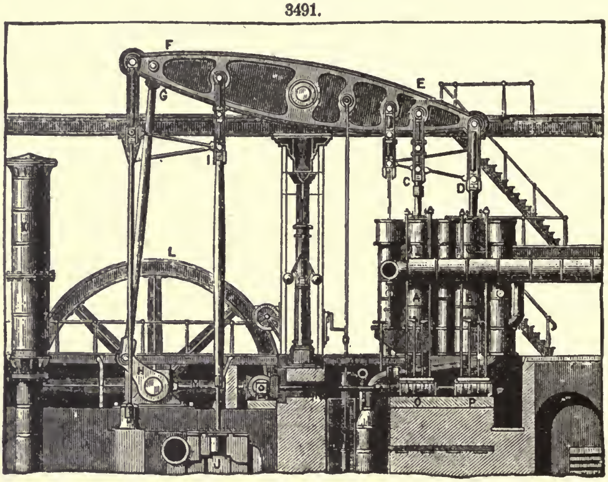


Fig. 9.—Simpson's Pumping Engine, 1848.

per 100 pounds of coal consumed on the grates of ordinary return tubular boilers, according to well authenticated reports.

About 1860 the Worthington duplex pumping engine began to appear and marked an era in pumping engine design. Taking up the direct acting or non-rotative idea where the Cornish engine had reached its limit as to size, economy, and adaptation to water works conditions, it carried on the march of improvement in a manner and to a degree which is strongly felt even to-day, nearly half à century later. Instead of long pit rods or heavy weights, the Worthington engine applies the force of steam during both strokes directly to the work to de done by

the plungers, by means of direct and rigid connections between the steam pistons and the water plungers. It was at first built only of the horizontal class, and up to what would now be considered very moderate sizes and capacities, but later entered the vertical field of design, until to-day it reaches to 25,000,000 U. S. gallons daily capacity and upwards of the vertical class. The early Worthington engines of fairly good size gave an economic duty of 60,000,000 foot pounds per 1,000 pounds of steam; and the latter day examples with high duty attachment develop a duty of nearly 175,000,000 foot pounds under favorable conditions.

Contemporary with the introduction of the Worthington duplex pumping engine there appeared the Holly quadruplex pumping engine, a crank and fly wheel engine of peculiar design, and this engine enjoyed an extensive application to water works service. In fact the Worthington and Holly engines monopolized and divided the water works business for many years to the exclusion of nearly all others, and were the forerunners of the present commercial pumping engines, the many repetitions of their manufacture producing them at a price so far below that of special engines designed for much higher steam economy, that the slow education of the public along really economic lines, and the improvements in machine shop management, were necessary for the substantial steps forward which we now view so complacently. The Holly quadruplex pumping engine gave a duty of about 85,000,000 foot pounds and if the clearances or waste room could have been reduced in a practical manner, the duty could no doubt have been considerably improved.

Early in the eighties the competition for supremacy and business between the two leading types of water works pumping engines, brought about a sharp line of demarcation between the low duty engines at low cost and a higher duty engine at a higher cost, resulting in the production of the Gaskill engine now known as the Holly-Gaskill or the Gaskill-Holly engine; and then the Worthington high duty engine. Both of these engines developed naturally enough along the previous lines of practice of their

respective builders, the Gaskill engine retaining the four cylinder of the quadruplex, although differently arranged, the new arrangement approaching somewhat that of its principal competitor, the Worthington, inasmuch as it embraced four horizontal steam cylinders and two horizontal double acting pumps, the difference being that instead of the tandem arrangement the Gaskill took on the form of the Woolf beam type of a hundred years ago, but horizontal instead of vertical. Many of the features of the valve gear of its predecessor, the quadruplex, were prominent in the Gaskill design. Its reported economic duty of 117,936,698 ft. lbs. per 1,000 lbs. of steam at Saratoga, in 1884, was much criticised and doubted at the time, just as the announcement of most improvements is so treated, but the value of small clearances or waste room in the ends of the steam cylinders was not then recognized so clearly and fully as later on. The writer had every reason to believe from his own experience that 118,000,000 ft. lbs. duty with compound pumping engines figured closely for steam clearance is a reasonable record for moderate sized engines built on commercial lines. The Gaskill pumping engine enjoys the distinction of being the first crank and fly wheel, high duty pumping engine, to be standardized and regularly built for water works service.

The Worthington engine was developed for high duty upon the same general plan as formerly constructed, with its two high pressure and its two low pressure cylinders, two double acting water plungers, and other features usual in the earlier practice. The economic duty credited to the Worthington high duty pumping engine was just about the same as that of the Gaskill and other engines of the same multiple expansion scope, viz: 120,000,000 duty per 1,000 lbs. of steam, or, as it was then the custom to put it, per 100 lbs. of coal, upon the supposition that the boiler work would be 10 lbs. of steam produced per pound of coal burned upon the grates.

The reasonableness of the claims for a compound condensing pumping engine, with cut off and expansion of steam, and with steam cylinders having a ratio from high to low pressure of

about four to one, being able to show a duty of about 120,000,000 ft. lbs. per 1,000 lbs. of steam, so far as the writer is concerned, was based at the time upon his personal experience and as time has passed there has been no need to adversely change views upon the subject; in fact, further refinements have raised the economic duty of the compound pumping engine somewhat above the 120,000,000 mark. The writer made a duty test with a Reynolds vertical compound condensing pumping engine at Hannibal, Mo., November 5, 1885, and upon the basis of 100 lbs. of coal and 10 lbs. evaporation, or really a basis of 1,000 lbs. of steam, obtained a duty of the then unprecedented figures under similar conditions of 118,327,041 ft. lbs., and considering the fact that the average steam pressure was only 79 lbs. per gauge, it is fair to assume that a somewhat higher pressure and a somewhat shorter cut-off in the high pressure cylinder, would have raised the duty a point or two above 120,000,000 ft. lbs. This engine was operated under regular everyday conditions and the feed water supplied to the boilers without deductions or allowances of any kind was taken as the steam consumed. The capacity of the engine per 24 hours was 2,826,056 U. S. gallons of 231 cubic inches; the total head against the plungers was 249 feet, and the piston and plunger speed was 165 feet per minute. The steam jackets and reheater consumed 10.6 per cent of the total steam supplied to the engine during the test. The dimensions of this engine are as follows:

One high pressure cylinder, diameter, 23 inches.
One low pressure cylinder, diameter, 45 inches.
Water plungers, each, diameter, 17 inches.
Stroke of all pistons and plungers, 30 inches.
Ratio of cylinders to each other, 3.98 to 1.
Volume of expansion during test, 15.03 to 1.
Piston and plunger speed, per minute, 165 feet.
Average water pressure, total load, 108.12 pounds.
Average steam pressure, gauge, 79 pounds.
Receiver of same volume as low pressure cylinder.
Two, single acting, outside packed plungers.

The Worthington high duty engine entered the field about the same time as the Gaskill engine, and these two types of pumping machinery for municipal water supply, represented two entirely different schools of principle in construction, although their heat theories were alike. The Worthington advocated the old Cornish principle of non-rotation; while the Gaskill set forth as earnestly the advantages of the crank and fly wheel school of practice; and these two engines came probably as near to commercial lines in their early days as the machine shop tools and practices of their time would permit, considering also the more or less special features of adaptability of any pumping engine to any particularly designated plant.

CHAPTER IV

THE ADVENT OF TRIPLE EXPANSION

In 1883 and 1884 the Reynolds pumping engine began to develop towards a commercial type, the one at Hannibal, Mo., referred to in the last chapter, being one of the earlier specimens which embodied features afterward retained, principally among them being the steam valves located across the cylinder heads, thus reducing the clearances or waste room to a very small percentage of the cylinder volume; the outside packed plungers; and the vertical type of machine. One of the early developments, after the Hannibal engine, was the vertical triple expansion for the high service station at Milwaukee, Wis., the first triple expansion pumping engine produced in the world, appearing in 1886 and designed to take the place of an engine similar to one of the Hannibal type. In 1892 this triple type of pumping engine broke all records for economic duty by sending the figures up to 154,048,700 ft. lbs. per 1,000 lbs. of steam, at the North Point pumping station of the Milwaukee water works.

Higher steam pressures and higher ratios of steam expansion, with the terminal pressure or the pressure at the end of the expansion, remaining at any certain point, are the essential features of a higher economic duty. A clear idea of the relation between expansion and steam economy may be grasped by considering that the terminal pressure represents the quantity of steam used, while the mean effective pressure or the average pressure throughout the stroke, represents the work done by the steam. Therefore it follows that the greater the mean effective pressure in proportion to the terminal pressure, the greater will be the economy of steam, and it also follows that the greater the rate or ratio of expansion with any certain terminal pressure, the

higher will be the duty of the engine, provided of course that proper conditions for the expanding steam are obtained. To show this in a convenient manner the following tables have been prepared in which three different terminal pressures are taken, and the mean effective pressures calculated for 12 different steam pressures per gauge, ranging from 80 lbs. to 150 lbs. per square inch.

But mere comparison between the terminal and mean effective pressures will not tell the whole story, because the lower the terminal pressure, the less density there is to the steam, or the less evaporated water per cubic foot of the steam. As, for example, the ratio between the terminal of 5 lbs. absolute, and the mean effective pressure for 33 expansions is 4 to 1 with 150 gauge pressure. While at the other extreme of the tables, the ratio between the terminal of 10 lbs. absolute and the mean effective pressure for 9.5 expansions is 3 to 1 with 80 lbs. pressure. But the steam at 10 lbs. pressure has nearly twice the density of the steam at 5 lbs. pressure, the weights per cubic foot being:

.02641 of a pound at 10 pounds pressure, per cubic foot.
.01373 of a pound at 5 pounds pressure, per cubic foot.

With a low pressure cylinder having say 3,000 square inches of piston area, or a little over 60 inches diameter, at 200 feet per minute travel of piston, with all steam accounted for by the indicator diagram, with no steam used in the jackets or reheater, and with 2.48 lbs. deducted for the best practical vacuum, the duty would be as follows:

Gauge Pressure.	Absolute Terminal.	Mean Effecvive Pressure.	Ratio of Expansion.	Duty per 1,000 Lbs. of Steam.
80	10	30.03	9.5	162,324,324
150	5	20.00	33.0	206,898,103

This shows a gain in economic duty of 27 per cent without allowing any steam to be used in the jackets; with an engine as

actually operated, and with steam used in the jackets and reheaters, the difference in actual economy was much more than this, between the first triple expansion pumping engine, and the engine holding the record at the end of the year 1900, but of course some allowance must be made for the practical improvements in pumping engines between 1886 and 1900, although from 1900 to 1906 the gain in duty was only 0.89 of one per cent. The difference in duty between 1886 and 1900 was as follows:

First triple in 1886, round numbers, 129,000,000 duty.
The record in 1900, round numbers, 179,000,000 duty.

This shows a gain of 38 per cent in actual practice and with the gauge steam pressure raised only from 80 lbs. to 126 lbs. But the later engine had poppet valves on the low pressure cylinder, thereby reducing the clearance to probably one half per cent or lower.

The high duty record from 1893 to April, 1906, all held at the various periods by the vertical triple expansion, crank, and fly wheel pumping engine using steam jackets and reheaters, is as follows:

Year.	Ft. Lbs. per 1,000 Lbs. of Steam.
1893	154,048,700
1895	157,843,000
1898	167,800,000
1900	168,532,800
1900	178,497,000
1900	179,419,600
1906	181,068,605

Following in tabulated form may be seen the ratios of expansion possible with different gauge pressures, and with different terminal pressures. The terminal and mean effective pressures being absolute or above perfect vacuum, and the expansion under the various pressures assumed to be as good as would be indicated by the Mariotte curve.

First: With a terminal of 5 pounds absolute.

Gauge Pressure.	Mean Effective Pressure.	Expansions.
80	19.72	19
90	20.02	21
100	20.67	23
110	21.18	25
115	21.29	26
120	21.47	27
125	21.66	28
130	21.84	29
135	22.00	30
140	22.16	31
145	22.32	32
150	22.48	33

Second: With a terminal of 7 pounds absolute.

Gauge Pressure.	Mean Effective Pressure.	Expansions.
80	23.37	13.5
90	25.96	15
100	26.64	16.4
110	27.15	17.8
115	27.49	18.5
120	27.67	19.3
125	27.98	20
130	28.13	20.7
135	28.35	21.4
140	28.67	22.1
145	28.80	22.8
150	29.18	23.5

Third: With a terminal of 10 pounds absolute.

Gauge Pressure.	Mean Effective Pressure.	Expansions.
80	32.50	9.5
90	33.50	10.5
100	34.41	11.5
110	35.26	12.5
115	35.65	13
120	36.02	13.5
125	36.39	14
130	36.74	14.5
135	37.08	15
140	37.40	15.5
145	37.72	16
150	38.00	16.5

These mean effective or average working pressures throughout the stroke of the piston are absolute pressures or pressures above an absolute and perfect vacuum; and the proportion of percentage of these pressures possible to actually obtain for useful work within a steam cylinder, will depend of course upon the value or perfection of the vacuum maintained within the low pressure cylinder of a triple expansion engine; it being considered quite out of the question to use one cylinder with a condenser, in economically carrying such high ratios of expansion, even with the lowest initial pressure given in the gauge pressure column of these tables.

During the past 25 years there has probably been nothing concerning the capabilities of pumping machinery so much discussed as its economic duty; or, the ability or inability to develop more or less foot pounds of work upon one basis or another of comparison, but mostly per 1,000 lbs. of steam consumed, or per 1,000,000 heat units utilized. Twenty-five years ago 60,000,000 foot pounds duty was the general guarantee for water works engines, with occasionally an engine of special design aiming to show 100,000,000 foot pounds, the advocates of the higher type and more costly machine arguing that the saving in fuel represented by the greater economy would more than pay the interest upon the difference in the cost of the machinery. This saving was a fact, upon the showing of the test and trial runs, provided the engines were equal in other respects, principally as to the ability to successfully and continually pump water without unusual interruption on account of stoppage, breakdown, and needed repairs.

Years ago the writer took the ground that the first duty of a pumping engine was to pump water, and sees no reason to change that idea to-day; but in the early days as a general rule the pumping engines of a higher duty than about 60,000,000 foot pounds, were not built quite so sturdy for the hard work of pumping, as their lower duty competitors, and if durability, smoothness of operation, and decreased cost had not gradually forged to the front, in the construction of pumping machinery,

for water works, the present admirable results would not have been attained.

The expansion of the steam is the key of course to higher and higher economy, but as this factor was increased in ratio from time to time and gradually approached its physical limits, new difficulties arose one after another, or were more and more realized as greater attempts at higher economy were made; but after persistent efforts along encouraging lines the happy compromise seems to at last have been reached between a highly elastic gas at one end of a machine, and a non-elastic most stubborn fluid at the other end, and with the limit of steam economy very nearly reached. In the early days of steam expansion to high ratios, it did not at first seem to be realized that the impact upon a steam piston of a very high initial pressure as against the corresponding effect of a low pressure, would greatly increase the shocks and possible damage to the machine. Under certain conditions, as in mill engines, the initial shock of the steam may be absorbed by adjusting the weight and motion of the moving parts to meet the sudden impulse of the incoming steam as the steam or induction valve is quickly opened; the piston traveling toward the end of its stroke must of course stop before it can make a return stroke, and although the unaided eye cannot determine the instant of the stoppage, it nevertheless actually takes place for the simple reason that it is physically impossible for it to go in both directions at the same time.

But having once stopped, or come to rest even if only for the shortest imaginable fraction of a second, it must be started again, and if the weight of the moving parts such as the piston rod, piston, crosshead, and connecting rod, can be made to just about equal the effort of the initial steam then the shock is taken up by the inertia of the mass. The speed of the crank pin in its circle or orbit of revolution, is practically the same at all points, but the driving parts of the engine moving only along the line across the circle, must of course start from no motion at the beginning of the stroke, come up to full speed of the crank pin at the middle

of the stroke, and then subside to no motion again at the end of the stroke. This means that the speed of the moving parts must be accelerated or increased and retarded or diminished twice at each revolution of the crank shaft. And the effect of this is precisely the same as that of centrifugal force at the beginning and end of the stroke, the calculation for the force required to start the weight at the beginning of the stroke being the same as that for determining centrifugal force which the weight exerts in trying to pull away from the center when rapidly revolved. If a plumb bob be attached to a piece of cord, say 6 feet long, and swung around in a horizontal circle just above one's head, the pull or pressure exerted by the centrifugal force, or the force due to the revolution of a weight, can be easily felt by the hand, and this pull or force is precisely what the steam will meet at the beginning of the stroke, so that if the force can be nearly or practically balanced against the incoming steam pressure, then smoothness and absence of wearing shock will be the result. In the latter half of the stroke the work represented by the rapidly moving weight brought up to the full speed of the revolving crank pin at each stroke, is utilized in helping to push the crank pin along its way until the end is again reached and the incoming steam again takes up the work for another stroke. Thus the first half of the stroke stores up the energy of the moving parts, which is again given out by these parts during the last half of the stroke.

This initial force is easily calculated from the fact that many experiments for the determination of how much centrifugal force amounts to, has demonstrated that one pound weight at the end of a crank one foot long, and making one revolution per minute, will exert a force pulling away from the center, of 0.000341 of a pound, or thirty-four thousandths of one per cent of a pound nearly. The relation of this amount of force to other conditions involving other weights, other lengths of crank, and other rates of revolution has resulted in the establishing of a formula or rule of calculation to the effect that the centrifugal force will vary in accordance with the following:

It will vary directly according to the weight in pounds.

Directly according to the length of the crank in feet.

According to the square of the revolutions per minute.

NOTE. — The square of the revolutions per minute, means the number of revolutions per minute, multiplied by itself.

The expression of the formula is:

Centrifugal force $= W \times R^2 \times L \times .000341.$

W represents the weight in pounds.

R^2 represents revolutions per minute multiplied by itself.

L represents the length of the crank in feet.

.000341 represents the centrifugal force of one pound making one revolution per minute at the end of a crank one foot long.

To show how this force will apply in practice take, for example, a 30-inch steam cylinder working condensing, indicating 40 lbs. mean effective pressure throughout the stroke; in one case using 100 lbs. steam pressure with 8 expansions, the initial load upon the piston would be 78,512 lbs., while in another case using 70 lbs. steam pressure in a throttling engine getting the same mean effective, the initial load upon the piston would be about 43,000 lbs.

With 100 lbs. initial pressure and 10 expansions the mean effective pressure would be 33 lbs. net, with an initial load of 78,512 lbs. as before, but the initial piston load upon the throttling engine would be 35,300 lbs.

With 125 lbs. initial pressure and 15 expansions the mean effective pressure would be 30 lbs. net, with an initial load of 93,722 lbs., while the initial load upon the piston of a corresponding throttling engine would be only 31,770 lbs.

This enormous difference in impact of initial pressures, coupled with the great amount of internal condensation within the steam cylinder when too high a rate of expansion was attempted in one cylinder, had a powerful effect in bringing about the very great extent to which load distribution is now made throughout the machine as a purely mechanical problem; also the thermal

idea of dividing the expansion into different stages and so reduce the range of temperature in each of the cylinders employed, more fully referred to in Chapter V, which deals with the Mariotte curve of expansion.

The adaptation of the pumping engine to its conditions and surroundings so far as may be, is the real key to highest efficiency under the particular conditions imposed. There are advocates of high speed, of high steam pressure, of high rate of revolution, and other singled out and isolated factors, but the general combination wherein the machine best meets the conditions is what will yield the best results, and not the exploiting of any particular seemingly important factor by itself. And, as an example of this fact, it may be noted that the pumping engine in this country, if not in the world, which up to April of 1906 held the high duty record has the following conditions to work under:

Capacity per 24 hours, 15,000,000 U. S. gallons.
Piston speed, 197 feet per minute.
Rotative speed, 16.41 revolutions per minute.
Water load against plungers, 126 pounds pressure.
Steam pressure per gauge, 126 pounds pressure.
Energy of steam end, 802 indicated horse power.
Energy of water end, 770 horse power.
Mechanical efficiency of the machine, 96 per cent.
Steam per hour per indicated horse power, 10.68 pounds.
Duty per 1,000 pounds of steam, 179,454,250 ft. lbs.

Since the above record was made in 1900 it has remained at the top of the list until April of 1906, when a similar engine newly installed in the same plant raised it to 181,068,605, or a gain of 0.89 of one per cent, which again emphasizes the idea that the limit is very nearly if not quite reached. This last record is so slightly in excess of its immediate predecessor that it requires several repetitions before it can be accepted as final, the error, cleverness, and good fortune attending some of these performances requiring at least one per cent leeway.

Regarding high piston speed, the best record known to the writer as to pumping engines is 607 feet per minute, where the

duty per 1,000 lbs. of steam was 157,843,000 ft. lbs., showing that higher piston speed alone will not answer the purpose.

Regarding high steam pressure, the record seems to be 200 lbs. and a duty of 149,500,000 ft. lbs., showing that high steam pressure in the absence of other ruling conditions or proper fitness falls short of the best performance.

Regarding thermal efficiency, or the actual economy of the heat employed, with reference to absolute temperatures, even the greatest thermal efficiency does not in the presence of adverse conditions in some other direction, equal the engine working under the best general fitness of things as will be seen by the following taken from the records:

Thermal Efficiency.	Duty per 1,000 Pounds of Steam.
Per Cent.	Foot Pounds.
22.80	149,500,000
21.63	178,497,000
21.00	179,454,250
20.85	173,620,000
20.78	176,419,600

What was considered the record up to April, 1906, for general all around efficiency for a pumping engine, and may yet be even after the particulars of the new record maker of 181,061,605 ft. lbs. per 1,000 lbs. of steam become known, is as follows:

Capacity per 24 hours, 30,000,000 U. S. gallons.

Steam pressure per gauge, 185 pounds.

Piston speed, 195 feet per minute.

Duty per 1,000 lbs. of steam, 178,497,000 ft. lbs.

Duty per million heat units, 163,925,300 ft. lbs.

Steam per indicated horse power hour, 10.335 pounds.

Thermal efficiency, 21.63 per cent.

The highest record for thermal efficiency is 22.80 per cent, which is 5.4 per cent above this all around record holder; but the latter shows a duty per 1,000 lbs. of steam which is 19.39 per cent above the former.

Up to April, 1906, the highest record for duty per 1,000 lbs. of steam was 179,454,250 ft. lbs. which is only 0.54 of one per cent above the all around machine, but the latter shows a thermal efficiency of 3.1 per cent above the former.

With reference to the steam economy of these higher types of pumping engines and its repetition in different engines, it may be noted that covering a period of more than 5 years, five pumping engines of a similar type, by different builders, and situated many miles apart, gave steam per indicated horse power ranging through the following figures:

10.33 pounds per horse power hour.
10.63 pounds per horse power hour.
10.78 pounds per horse power hour.
11.01 pounds per horse power hour.
11.10 pounds per horse power hour.

As already pointed out, it hardly seems probable that materially higher efficiencies will be obtained in the near future, and not very much higher duties are possible with the steam pressures seemingly practicable to employ in pumping stations, in fact, so far at least, the use of quadruple expansions and 200 lbs. steam pressure has not resulted in so good work as the triple expansion with 126 lbs. pressure. Indeed the employment of 250 lbs. steam pressure and 50 expansions does not promise even theoretically very much gain; and practically, in large pumping units available practice is against such an advance in the difficult line of high expansion. Superheat will no doubt carry the results to slightly higher figures than at present, but the conditions must be very carefully met to make it profitable to the owner and user. Net gain is what is sought, and fancy duty figures at the expense of heat and repairs in some other part of the plant will not add to the real economy in the long run. But however the steam or heat efficiency may be improved, or however many times the present record may be reached in the future, the durability of the machinery for the purpose of its existence comes first and should not be sacrificed

to any fancied betterment in steam economy only. The following table shows the economical efficiency of the higher types of pumping engines produced from 1893 to 1903 and includes about all of the prominent builders in this country.

Date of Record of Engine Test.	Capacity in Gallons per 24 Hours.	Piston Speed Feet per Minute.	Total Water Load Pounds Pressure.	Gauge Steam Pressure in Pounds.	Thermal Efficiency in per cent.	Duty per 1,000 Pounds Steam.
1893	18,000,000	203	71	121	19.40	154,048,700
1894	16,000,000	371	84	137	19.07	148,655,000
1895	20,000,000	607	60	176	20.76	157,843,000
1897	30,000,000	208	86	167	18.35	152,000,000
1898	20,000,000	215	89	156	20.45	167,800,000
1899	20,000,000	200	88	149	19.90	165,220,000
1899	6,000,000	256	262	200	22.80	149,500,000
1899	10,000,000	175	128	136	18.44	160,455,000
1899	10,000,000	175	128	137	18.59	161,530,000
1900	12,000,000	211	115	173	20.00	168,532,000
1900	15,000,000	198	127	127	20.78	176,419,600
1900	15,000,000	197	126	126	21.00	179,454,250
1900	30,000,000	195	61	185	21.63	178,497,000
1901	21,000,000	496	82	178	19.43	146,173,000
1901	35,000,000	300	20	151	20.50	157,349,000
1901	20,000,000	248	54	150	20.85	173,620,000
1903	10,000,000	480	80	181	18.95	140,000,000
1903	15,000,000	197	127	138	20.72	177,300,000
1903	15,000,000	197	127	135	20.67	177,200,000

In the above table the first engine is a vertical triple; the second one a vertical compound; the third, fourth, fifth, and sixth are vertical triples; the seventh is a quadruple; and the rest, with the exception of the seventeenth which is a vertical compound, are vertical triples.

CHAPTER V

THE MARIOTTE CURVE

The Mariotte law of expanding gases was discovered by Edme Mariotte, Prior of St. Martin, and one of the first members of the Academy of Sciences which was founded at Paris, France, in 1666. He was a native of Burgundy, and died in 1684 after a long and useful life.

This law was also discovered by Boyle in 1662, separately from and independently of Mariotte, and is sometimes called Boyle's law, especially in England, but on the Continent of Europe and in this country is mostly known as Mariotte's law of volume and pressure of gases. It is laid down in the reference books as follows:

The temperature remaining the same, the volume of a given quantity of gas is inversely as the pressure which it bears.

That is to say, when the temperature of a gas does not change from a higher to a lower degree, or the reverse, if the volume or cubic contents is reduced to one half of what it was formerly, its pressure will be doubled. Or, if the gas under the same conditions of temperature is increased in volume so as to fill twice the space as before, its pressure will be reduced to one half. And this effect will follow any other proportionate change in volume; four times the volume makes one fourth the pressure; one fourth the volume makes four times the pressure; and so on to five, to six or any other relative change apparently up to twenty-seven at least and that is as far as experiments were carried of a similar nature.

This law has been considered and experimented with by different investigators and found to be correct up to at least 400 lbs. pressure per square inch, so we may consider it conclusive

so far as it fits the case, and so far as ordinary useful pressures for pumping engines are concerned. The law can be very easily demonstrated by a glass tube bent like a letter U with the open ends of the tube uppermost and with the straight parts of the U vertical. One leg of the U is much shorter than the other, and near the lower end of the U close to the bend there is marked a level line and up to this line, in both legs, a small quantity of mercury is poured, making the surface of the mercury in both legs alike in level. By stopping up the top of the short leg and pouring additional mercury into the long leg, the result will be that when the body of imprisoned air has been reduced to half its original volume at the closed end of the glass, the height of the mercury will show that the pressure is doubled. The reverse can be shown by reversing the tube so that it will be inverted from its original position, and then the mercury will indicate half the pressure when the volume of the original air has been doubled.

The law provides that the temperature remain the same, and this provision can be satisfied in the glass tube experiment; but with cylinders and pistons doing work by the expansion of a gas or fluid, the temperature of the air or any other gas would not remain the same, but would cool by expansion and heat by compression.

The writer had occasion several years ago to examine and report upon a large air compressing and operating plant at Iron Mountain, Mich. The object of this plant was to compress air by means of water power at Menominee Falls, and send the compressed air under 60 lbs. pressure four miles through a 24-inch steel main to the Chapin and Ludington iron ore mines for operating, pumping, drilling, hoisting, and other kinds of mining machinery. The air compressor cylinders were 32 inches diameter and of 60 inches stroke of pistons; in compressing the free air up to 60 lbs. pressure an indicator diagram showed that about 20 per cent was added to the area of the diagram by the heating of the air up to from 250 to 300 degrees Fahrenheit. This heat was represented by 772 foot pounds for each amount of the heat that would raise one pound of water one degree, from 39

degrees Fahrenheit. At the mine, four miles from the falls, the air had cooled down to atmospheric temperature, where a large bob pump was worked by a Reynolds-Corliss engine, 24 by 48, at 50 revolutions per minute, and the indicator cards showed a shrinkage in the diagrams of about 15 per cent at the expansion curve, expanding from 60 initial down to the ordinary atmospheric pressure. In the absence of a small steam jet which was kept going in the exhaust chamber of the engine cylinder, and which was shut off now and then for experimenting, frost would form on the back cylinder head with the sun shining full upon it; the air remaining perfect air, of course, during these operations of absorbing and giving out heat, because air is a perfect gas and the heat has no part in its composition.

However, steam is not a perfect gas, like air, oxygen, hydrogen, and other gases. Steam is a mixture of heat and water, and those two elements quickly separate when left to themselves without additional heat to make up for the inevitable losses from radiation or work. Consequently, in an actual engine cylinder, and doing work, marked disturbances take place; and this imitation gas which we call steam suffers at the beginning of the stroke and recuperates, and a little more, at the latter portion of the stroke; the general result on an indicator card in very good practice being a pretty close copy of the Mariotte expansion curve by a perfect gas; and, if we treat the expansion, quantity of steam, and the work done, by steam in an engine, strictly according to the form of the Mariotte curve, it will be seen that the results arrived at are very desirable and hard to attain in the actual practice with steam engines.

It is a case of a balancing of errors, and gives the advantage of showing what may be done by what might be called a reflected light. But the reflected light shows the road just the same. The variation in conditions of actual steam, heat, and work, are so many, the fluctuations of values in the materials used are so great and so incessant, that it is absolutely impossible to lay out in advance a scientific sharp line which can be hewn to. And

therefore to the writer it seems to be much better to have a hard and fast line to go by, amendable to mathematical precision, and capable of being produced on every and on all occasions for purposes of seeing just where we are at any time. In short, if the engineer sets up the Mariotte curve and law, and assumes perfectly dry steam, he will have a much better light to look forward by, than he will by endeavoring to account for all the variations which he will surely meet in trying to follow all of the actual facts; and he will find that considering the actual consumption of steam and fuel, he will get better results in pumping water. Not only that, he will find in working out his salvation by the Mariotte curve, that he will approximate very closely indeed to proportions and dimensions which produce the highest results in actual steam practice, as will be shown further along in this book. There are other curves in the expansion diagram, carrying the line a trifle below and inside of the Mariotte curve, requiring a deeper look into the science of thermodynamics than the writer deems desirable in a book of this scope. But such curves, although necessary to follow and plot out in obtaining a good theoretical grasp of the subject, can never be normally made by an indicator attached to a steam engine; and, as the Mariotte line is the very best that can be made in practice without leakage or condensation, it is the one thought best to be used for the purposes of illustration.

Considering the foregoing then, the following examples are based upon the Mariotte expansion curve, perfectly dry steam, all working steam accounted for by the indicator diagrams, and allowing for no clearance or waste room in the steam cylinders. Indeed the waste room has been brought down to such a fine point, less than half of one per cent, in many cylinders, that it can practically be neglected as there are other disturbances of more consequence. The diagrams in these examples are made to conform to the Mariotte curve as the standard for comparison for all steam expansion curves. This curve is a mathematical one and easily determined; being a hyperbola readily laid out according to its law of relative volumes and pressures, and its

effects may be readily calculated by means of the simple rule and a table of hyperbolic logarithms. The Mariotte is about the best curve that can be accomplished in actual practice and in fact it is rather difficult in the absence of leakage or other undesirable conditions to get the expansion down even to this curve. The other curves mentioned above, determined by intricate theoretical and mathematical considerations, show the expansions of steam after all is accounted for as going a little lower than that of the Mariotte law, and therefore indicating

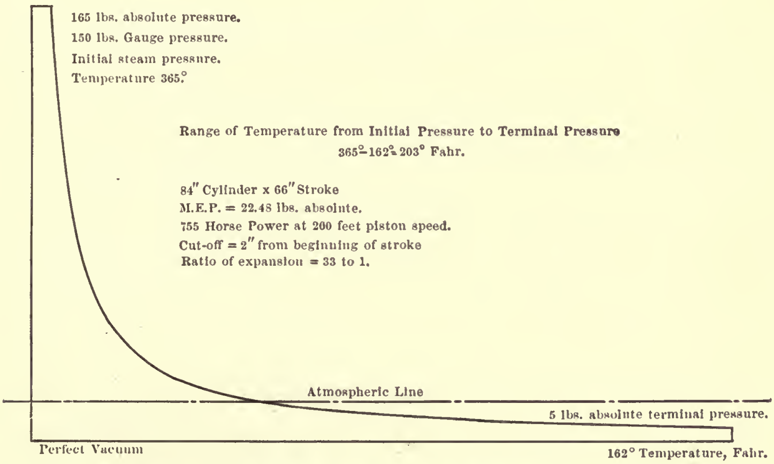


Fig. 10.—The Mariotte Curve in a Single Cylinder.

a trifle better economy of steam, if such curves could be honestly made by an indicator attached to a steam cylinder. But as there is very little difference after all, and as the very best curves made in practice essentially coincide with that of the Mariotte law, the diagrams used for these illustrations may be taken as practically correct; about the only remaining question being as to how far will the indicator account for the steam used, which after all leads to a further question as to the quality of the steam at the low pressure terminal, it being a pretty fair assumption that with the low pressure terminal perfectly dry, and not materially superheated, about all of the working steam

will be accounted for by the pressure shown and the volume expressed by the low pressure cylinder, at that portion of the sweep of its piston which carries it to the point of release of the exhaust steam.

First Example (See Fig. 10).

One steam cylinder, 84 inches diameter, 66 inches stroke.

Initial steam pressure 165 lbs. absolute, or 150 lbs. per gauge.

Terminal pressure 5 lbs. absolute, or above perfect vacuum.

Ratio of expansion, 33 to 1.

Point of cut-off, 2 inches from beginning of stroke, no clearance.

Mean effective pressure for the stroke, 22.48 lbs.

Area of the piston, 5,541 square inches.

Total mean force effective on the piston, 124,579 lbs.

Gross initial pressure on the piston, 914,265 lbs.

Range of temperature, initial to terminal pressures, 203 degrees Fahrenheit.

At 200 ft. per minute, 755 indicated horse power will be developed.

At 96 per cent efficiency of machine, 725 pump horse power would be shown.

Steam per hour per diagram would be	6,334 lbs.
Steam for jackets etc. per record, 15.45 per cent	1,158 lbs.
Total steam per hour	7,492 lbs.

Steam per indicated horse power hour, 9.92 lbs.

Steam per pump horse power hour, 10.33 lbs.

Duty per 1,000 lbs. of steam, 191,674,830 ft. lbs.

An engine such as outlined above would be impracticable if not impossible. The great range of temperature indicated between what steam would have at 165 lbs. initial pressure and 5 lbs. terminal pressure, could not be taken care of by a steam jacket quickly enough to protect the working steam during one stroke in any engine it would be practicable to build. And

further, the great initial load of 914,265 lbs. would be extremely difficult to accommodate in a profitable manner in an engine of the power indicated, and with this power to be distributed throughout the machine for the purpose of pumping water in an acceptable manner.

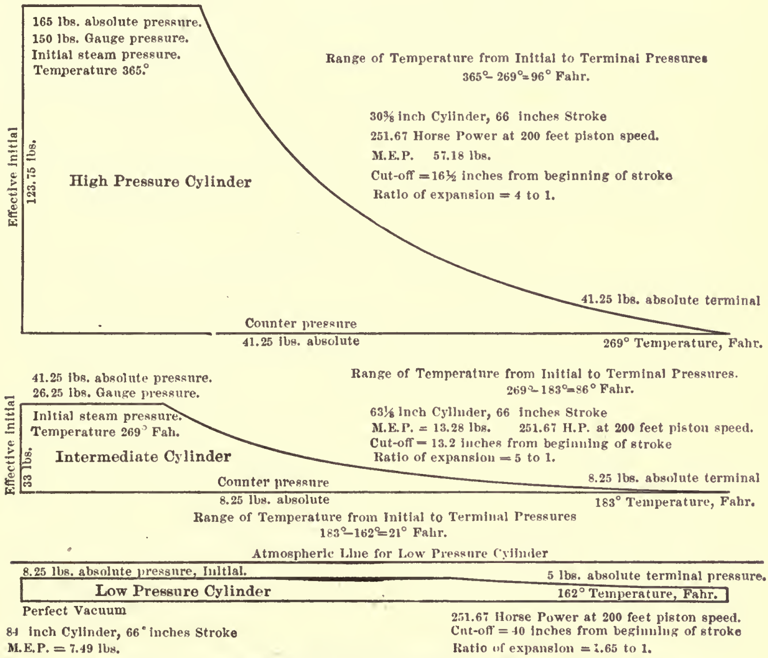


Fig. 11.—The Mariotte Curve Through Three Cylinders.

Second Example (See Fig. 11).

Triple Expansion, or the Same Work Through Three Cylinders.

High pressure cylinder, $30\frac{3}{8}$ inches diameter.
Intermediate cylinder, $63\frac{1}{8}$ inches diameter.
Low pressure cylinder, 84 inches diameter.
All of 66 inches stroke.
High pressure initial, 165 lbs. absolute.
High pressure terminal, 41.25 lbs. absolute.
High pressure effective initial, 123.75 lbs.

High pressure ratio of expansion, 4 to 1.

High pressure point of cut-off, 16.5 inches from beginning of stroke.

High pressure mean effective pressure, 57.18 lbs.

Area of the piston, 726 square inches.

Initial load on piston, 119,790 lbs.

Total mean effective force on piston, 41,526 lbs.

Range of temperature, initial to terminal pressures, 96 degrees.

At 200 ft. per minute, 251.67 horse power would be developed.

Intermediate cylinder initial, 41.25 lbs. absolute.

Intermediate terminal, 8.25 lbs. absolute.

Intermediate effective initial, 33 lbs.

Intermediate ratio of expansion, 5 to 1.

Intermediate point of cut-off, 13.2 inches from beginning of stroke.

Intermediate mean effective pressure, 13.28 lbs.

Area of the piston, 3,129 square inches.

Initial load on piston, 129,070 lbs.

Total mean effective force on piston, 41,526 lbs.

Range of temperature, initial to terminal pressures, 86 degrees.

At 200 ft. per minute, 251.67 horse power would be developed.

Low pressure initial, 8.25 lbs. absolute.

Low pressure terminal, 5 lbs. absolute.

Low pressure effective initial, 8.25 lbs. absolute.

Low pressure ratio of expansion, 1.65 to 1.

Low pressure point of cut-off, 40 inches from beginning of stroke.

Low pressure mean effective pressure, 7.49 lbs.

Area of the piston, 5,541 square inches.

Initial load on piston, 45,713 lbs.

Total mean effective force on piston, 41,526 lbs.

Range of temperature, initial to terminal pressures, 21 degrees.

At 200 ft. per minute, 251.67 horse power would be developed.

Sum of the power of all three cylinders, 755 indicated horse power.

At 96 per cent efficiency of machine, 725 pump horse power would be developed.

Steam per hour per diagram would be	6,334 lbs.
Steam jackets per record, 15.45 per cent . . .	1,158 lbs.
Total steam per hour	7,492 lbs.

Steam per indicated horse power hour, 9.92 lbs.

Steam per pump horse power hour, 10.33 lbs.

Duty per 1,000 lbs. of steam, 191,647,830 ft. lbs.

Initial load of all three pistons together, 249,573 lbs.

A comparison of these two examples will show the great reduction in the range of temperature in any of the cylinders of the triple engine; in the high pressure cylinder the range is only 96 degrees as compared with 203 degrees of the single cylinder condensing engine; in the intermediate cylinder the range of temperature is only 86 degrees, while in the low pressure cylinder, where the greatest damage to the steam would naturally be done, the range is reduced to the insignificant amount of 21 degrees, or only a little over 10 per cent of what it would be between the pressures due to the expansion in a single cylinder of equal dimensions. And, as in both cases the low pressure cylinder is next to the condenser, the importance of the change brought about by multiple expansion cylinders becomes apparent.

Further than this there are several incidental reasons why the three cylinders will give much better effects practically than one large cylinder with reference to the steam jacketing.

It will be noted that in the high pressure cylinder of the triple, the range of temperature due to the two pressures of steam, one at the initial of 165 lbs. and the other at the terminal of 41.25 lbs. both absolute pressures, is not only reduced to less than half of that in one large cylinder, between 165 and 5 absolute, but also the jacket surface in the high pressure cylinder amounts to 1.08 square feet of jacket surface per cubic foot of cylinder, including the area of one cylinder head, while the jacket surface is only

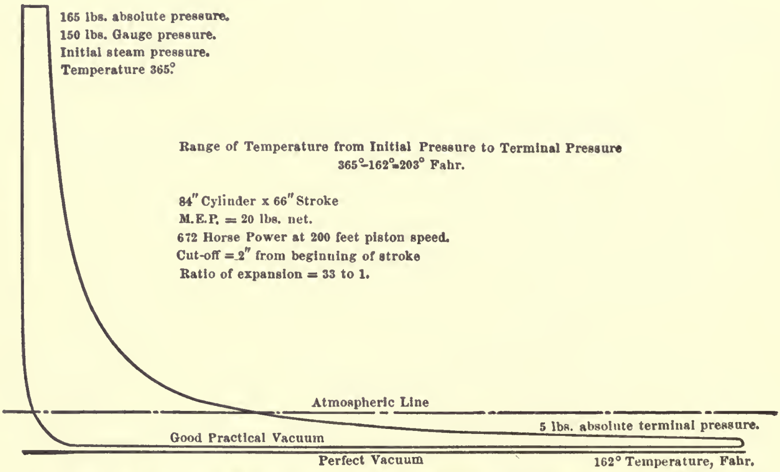


Fig. 12.—The Mariotte Curve in a Single Cylinder in Practice.

.75 of a square foot of heating surface for each cubic foot of cylinder for this one large cylinder.

This tabulated is as follows:

Jacket Surface per Cubic Foot Contents of Cylinder

84 inch cylinder, 0.75 sq. ft.
63 inch cylinder, 0.92 sq. ft.
30 inch cylinder, 1.80 sq. ft.

The lesson from the above is that the range of temperature represented by the sensible heat of steam at the initial and at

the terminal pressure, in the high pressure cylinder, is less than half of that in a single large cylinder. The ratio is 2.4 to 1 in favor of the high pressure cylinder, and the radiating distance from the side surfaces to reach the center is 2.8 times greater in the large cylinder than in the high pressure cylinder. The

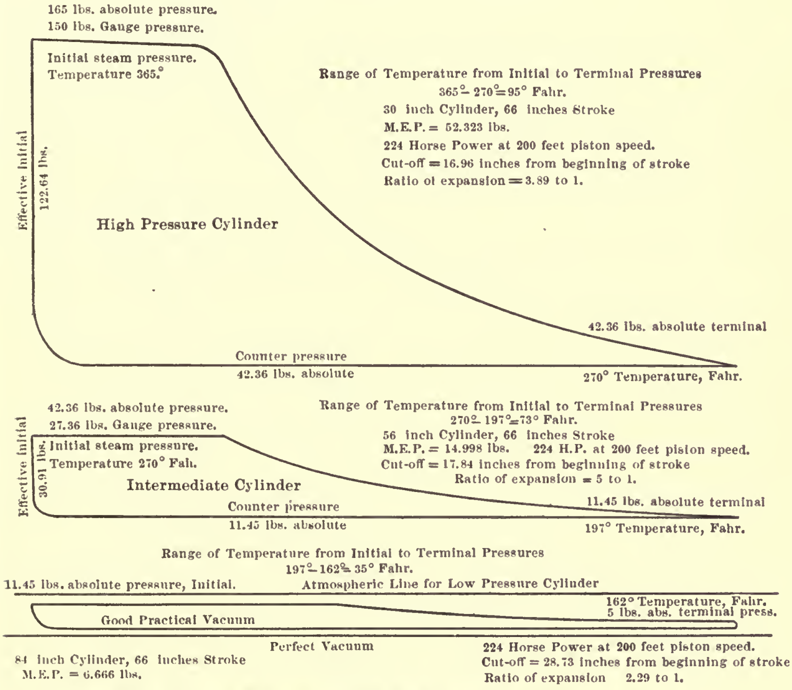


Fig. 13.—The Mariotte Curve in a Triple Engine in Practice.

range of temperature represented by the difference in initial and terminal pressures in the intermediate cylinder is less than half that in the large single cylinder and with a greater proportionate jacket area. And finally the range of temperature in the low pressure cylinder of the triple is only about one-tenth of that of the single cylinder, and with an equal amount of jacket surface.

The work done in the water end of a pumping engine under the

foregoing conditions would pump water as per the following table:

U. S. Gallons, 24 Hours.	Lbs. Pressure.
10,000,000	180
15,000,000	120
20,000,000	90
30,000,000	60

Third Example (See Fig. 12).

Single Cylinder, Straight Condensing Engine, in Practice.

One cylinder, 84 inches diameter, 66 inches stroke.
Initial steam pressure 165 lbs. absolute, or 150 lbs. per gauge.
Terminal pressure 5 lbs. absolute, or above perfect vacuum.
Ratio of expansion, 33 to 1.
Point of cut-off, 2 inches from beginning of stroke, no clearance.
Mean effective pressure for the stroke, 22.48 lbs.
Loss shown by best practicable vacuum, 2.48 lbs.
Net mean effective pressure in one 84 inch cylinder, 20 lbs.
Area of the piston, 5,541 square inches.
Total net mean effective force on the piston, 110,820 lbs.
Range of temperature from initial to terminal pressures, 203 degrees.
At 200 ft. per minute, 672 indicated horse power would be developed.
At 96 per cent efficiency of machine, 645 pump horse power would be developed.

Fourth Example (See Fig. 13).

Triple Expansion Pumping Engine in Practice.

High pressure cylinder, 30 inches diameter.
Intermediate cylinder, 56 inches diameter.
Low pressure cylinder, 84 inches diameter.
All of 66 inches stroke.

Total net piston force as per Third Example, 110,820 lbs.

One third of 110,820 lbs. each piston of triple engine, 36,940 lbs.

High pressure initial, 165 lbs. absolute.

High pressure terminal, 42.36 lbs. absolute.

High pressure effective initial, 122.64 lbs.

High pressure ratio of expansion, 3.89 to 1.

High pressure point of cut-off, 16.96 inches from beginning of stroke.

High pressure mean effective pressure, gross, 57.64 lbs.

Deficiency in practical high pressure diagram, 5.317 lbs.

Net mean effective pressure, high pressure cylinder, 52.323 lbs.

Area of the piston, 706 square inches.

Total mean effective force on high pressure piston, 36,940 lbs.

Range of temperature from initial to terminal pressures, 95 degrees.

At 200 ft. per minute, 224 horse power would be developed.

Intermediate cylinder, initial 42.36 lbs. absolute.

Intermediate terminal, 11.45 lbs. absolute.

Intermediate effective initial, 30.91 lbs.

Intermediate ratio of expansion, 3.7 to 1.

Intermediate point of cut-off, 17.84 inches from beginning of stroke.

Intermediate mean effective pressure, 14.998 lbs.

Area of the piston, 2,463 square inches.

Total mean effective force on piston, 36,940 lbs.

Range of temperature from initial to terminal pressures, 73 degrees.

At 200 ft. per minute, 224 horse power would be developed.

Low pressure initial, 11.45 lbs. absolute.

Loss shown by best practical vacuum, 2.48 lbs.

Low pressure effective initial, 8.97 lbs.

Low pressure terminal, 5 lbs. absolute.

Low pressure ratio of expansion, 2.29 to 1.

Low pressure point of cut-off, 28.73 inches from beginning of stroke.

Low pressure mean effective pressure, gross, 9.14 lbs. absolute.

Loss shown by best practicable vacuum, 2.4733 lbs. absolute.

Low pressure net mean effective pressure, 6.6667 lbs.

Area of the piston, 5,541 square inches.

Total mean effective force on piston, 36,940 lbs.

Range of temperature from initial to terminal pressures, 36 degrees.

At 200 ft. per minute, 224 horse power would be developed.

Sum of the powers of all three cylinders, 672 indicated horse power.

At 96 per cent efficiency of machine, 645 pump horse power would be shown.

Steam per hour per diagram would be	6,334 lbs.
Steam jackets per record.	1,158 lbs.
Total steam per hour	7,492 lbs.

Steam per indicated horse power per hour, 11.14 lbs.

Steam per pump horse power per hour, 11.67 lbs.

Duty per 1,000 lbs. of steam, 169,657,240 ft. lbs.

With 10% jacket consumption, the steam per indicated horse power per hour would be, 10.47 lbs.

And the duty per 1,000 lbs. of steam, 181,484,876 ft. lbs.

Note. — This demonstration was written about 7 months before the new record of 181,065,943 duty per 1,000 lbs. of steam was made at St. Louis in April, 1906.

The raise in duty between 15.45% and 10% of the total steam consumed in the jackets, shows the importance of limiting the use of steam in the steam jackets to the lowest needed amount, and some further remarks will be made upon this point under the head of steam jackets. To state this jacket percen-

tage the other way; with 15.45% of the total steam consumed in the jackets, 84.55% would be used as working steam in the cylinders; and with 10% consumed in the jackets, the working steam would amount to 90% of the total used; and as the amount of working steam would be about the same in any good case, it follows that the less jacket steam we can get along with, the less the total steam will be to charge against the work done by the engine. Then if by some marvel all the steam ordinarily used in the jackets could be saved and the working steam in the cylinders be kept dry, 100% of the total steam would be expressed by that accounted for by the diagram, which in the above case is 6,334 lbs. per hour. The statement then would be as follows:

Steam per indicated horse power per hour, 9.43 lbs.
Steam per pump horse power per hour, 9.82 lbs.
Duty per 1,000 lbs. of steam, 201,629,327 ft. lbs.

This apparently marks the limit for 150 lbs. gauge steam pressure and the Mariotte curve, and in this connection it may be of interest to note that some years ago, when 154,000,000 ft. lbs. duty marked the limit of accomplishment, an eminent professional authority remarked that this record was about 25% below the ideal results. And 201,629,327 discounted 25% amounts to 151,221,986 duty, which seems to indicate that the reflected light and the real light focus on the same point ery nearly.

Further than this, to show a still closer relation it may be observed that the latest large pumping engine, of the vertical triple, crank, and fly wheel type, designed for 150 lbs. gauge pressure has steam cylinders as follows:

High pressure cylinder, 30 inches diameter.
Intermediate cylinder, 56 inches diameter.
Low pressure cylinder, 84 inches diameter.
All with a stroke of 60 inches.
Capacity per 24 hours, 15,000,000 U. S. gallons.
Working load against plungers, 120 lbs.

The dimensions which work out in this chapter as an ideal engine, with which to consider the Mariotte curve, and supposed to be of the vertical triple expansion, crank and fly wheel type has steam cylinders as follows:

High pressure cylinder, 30⅜ inches diameter.
Intermediate cylinder, 63⅛ inches diameter.
Low pressure cylinder, 84 inches diameter.
All with a stroke of 66 inches.
Capacity, 15,000,000 U. S. gallons against 120 lbs. load.

CHAPTER VI

STEAM JACKETS

THE measure of economy brought about by the use of the steam jacket applied to the cylinders of steam engines has long been a disputed question. But in all such matters the evidence comes forward slowly on account of the many varying conditions involved, and the scarcity of opportunities for making demonstrations both ways, with the engine design, the adaptability of the design to the work, and other matters remaining the same. It is very easy to conceive of a large and important engine being badly adapted to its work; or, the steam jackets on any sized engine being badly proportioned or badly applied, or badly supplied with steam, either too much steam or too little. The saving of one pound of steam per hour with an engine of say 1,000 horse power, and this is not a large machine these days, means as per the following statement:

Steam per hour supposed to be saved by steam jacket, 1,000 lbs.
Coal at 8 lbs. evaporation, per hour, 125 lbs.
Fraction of a ton of 2,000 per 10 hours, .625.
Coal for 10 hours, 1,250 lbs.
Cost of .625 of a ton at $3 per ton, one day $1.87.
Cost of coal saved for 300 days per year, $562.50.
Capital upon which 5% would be earned, $11,250.

The main question is after all, will a part of the steam going through the jackets and a part going through the cylinders result in the use of less total steam than all of the steam going through the cylinders, or will it not?

The writer has experimented a good deal with steam jackets for the past 20 years, and distinctly remembers that upon several

occasions when the jacket steam had been greatly restricted for experimental purposes, and the record made by several hours run, the steam supply valve for the jacket was again opened up full on; whereupon the engine increased its speed two or three turns per minute. This being immediately noted and that no other condition had been changed, experimental runs were made, and, although a little greater speed was maintained with the increased amount of steam going through the jackets, and a little more water pumped, generally about 3 per cent, there was a net loss for the reason that the increased steam let into the jackets was more in proportion than the gain in mechanical work; the jacket supply was again cut down and the net efficiency rose at once. This demonstrated that too much steam could be admitted to a jacket and also that the heat will do the work whether it goes through the main throttle valve into the working cylinders, or through the metal of the cylinders. So far as coal consumption, and therefore the cost of the power from that item was concerned, it did not matter whether the heat went through the metal and after bringing the working steam up to the best point of efficiency, sent a surplus of heat into the exhaust and into the traps; or whether the cut-off was let out and steam followed the piston to a little later point in the stroke. It was also demonstrated that when the jacket valve was opened wider, the cut-off point had to be set back to keep the engine down to normal speed, and when the jacket steam was restricted the cut-off point had to be advanced to maintain the speed up to the mark. Therefore it follows that it makes no difference in gaining or reducing speed whether more or less steam is admitted up to cut-off or into the jackets, so far as each of these items is concerned separately. The proper combination of jacket and cut-off conditions however control the item of steam economy.

There are so far no claims that the steam jacket has made any losses in any plant at all adapted to the work in hand, and the records mostly show at least that the jackets even in the cases of apparently bad showing, pay for their own steam and a little more, but whether they pay for their existence seems to be in

some cases doubtful to say the least. The steam jacket is not a very costly addition to a steam cylinder and the net saving does not have to be so very much to pay for its being put there. A point not made entirely clear so far with mill engines, is whether or not the gross work, or the indicated horse power, is more or not without the jackets. All conclusions with mill engines have of course been based upon the indicated power, and in the case of a dry cylinder with jackets as against a damp cylinder without jackets, the former may furnish a greater proportionate amount of useful power; which means that with moist steam in the cylinder some perceptible effort may be necessary upon the part of the engine to overcome the "laziness" of the piston. In mill engines this is not an easy point to determine, but with a pumping engine wherein the net work is so easily measured, all or nearly all results favor the steam jacket. When Geo. H. Corliss made his five cylinder straight condensing pumping engine for the city of Providence, back in the seventies, his results would no doubt have been very much better had he fitted the engine with steam jackets, as is evidenced by the enormous evaporation and high terminal pressure in the steam cylinders.

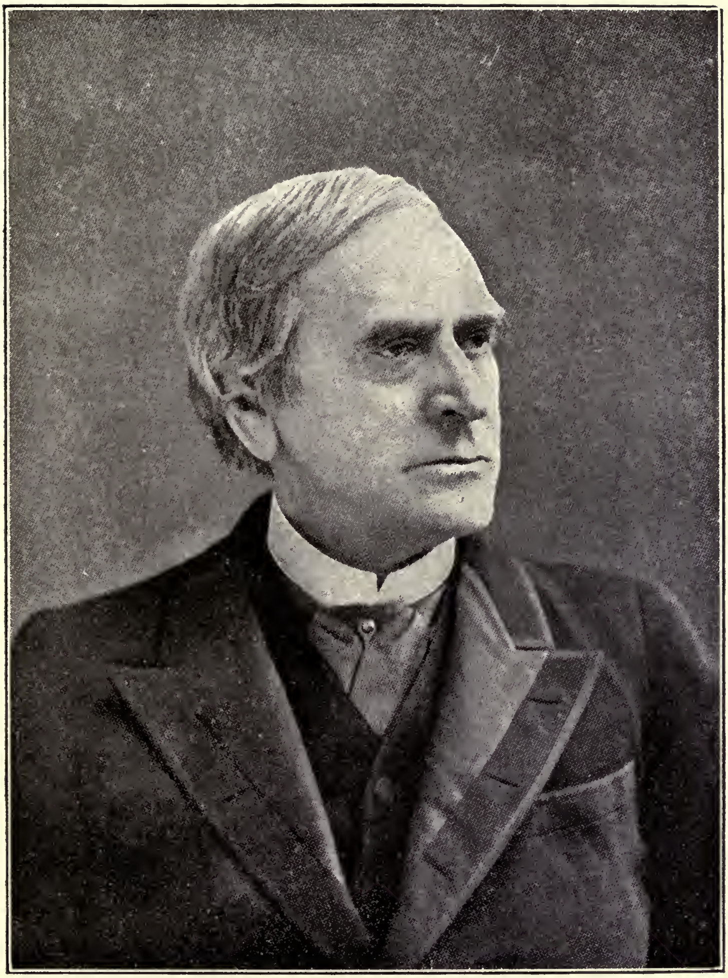
A case in point is as follows:

A compound duplex condensing pumping engine of 2,000,000 gallons daily capacity had been installed, and under the contract had been calculated to operate without steam jackets. But to save time, and the making of new cylinders, a set of cylinder castings was used which had been on hand for some time, and which had steam jackets. The engine was started up without connecting up the jackets, and, although the indicated horse power seemed to be ample for the water load, and the steam per indicated horse power seemed to be reasonable, the engine could not be brought up to speed. In other words the steam per horse power hour was all right but the apparent mechanical efficiency of the machine was all wrong. After testing out for line and for undue friction in the engine, and these found to be normal, the steam jackets were connected up and the trouble was overcome at once.

Where reheating effects are sought in chimney flues the apparent usefulness of the steam jacket may appear small; but if the heat of a chimney flue is applied to its legitimate work of heating feed water or is absorbed in making steam within the boiler before reaching the flue, the net results might be better than letting such heat escape from the boiler and then attempt to get it back again into steam on its passage from one cylinder to another by the use of coils in the boiler flue. The net profit of a steam jacket lies largely in the manner in which the jacket is adapted and applied.

In the modern pumping engine of high efficiency and of comparative high first cost, but which when properly designed and built, and properly adapted to its task, returns a very handsome profit upon its excess of cost over a cheaper but more extravagant competitor, it is sometimes impossible to use the high rates of expansion needed for desirable results without condensing from 25% to 40% of the initial steam within the cylinders in the absence of steam jackets. Therefore, if we can expend say 15% of the total steam and save 30% then the use of steam jackets will pay.

To illustrate this, supposing that a pumping engine could be supplied with 7,000 lbs. of steam per hour from the boilers; and that without steam jackets, say, 30% of the total steam will be condensed within the cylinders. Then we should have a loss of 2,100 lbs. of steam, making only 4,900 lbs. available for work per hour. This would at the best record give 472 horse power, and counting the lost steam there would be a consumption of 14.7 lbs. per horse power per hour without steam jackets. By the use of steam jackets in such an engine properly applied and operated, if 15% of the total steam supplied should be used in the jackets and most all of the initial condensation prevented, then the amount of steam saved per hour would be 1,050 lbs. giving available for work, 5,950 lbs. of steam, which at the record rate per horse power would develop 573 horse power, or give a horse power for 12.2 lbs. of total steam. The gain in power would be 21% and at 8 lbs. evaporation with coal at $3



GEORGE H. CORLISS.

per ton the difference in cost of power in favor of steam jackets would be 5% per annum on $34,492. Such perfection as mentioned above would be very hard to reach, but the saving by the jackets would without doubt reach the net amount in the steam of 10%, and this at 8 lbs. evaporation and with coal at $3 per ton would pay 5% per annum on $22,995, which represents a great deal more than the difference between jackets and no jackets in the matter of construction.

Therefore it can safely be said that with pumping engines, at least under most conditions, enough more steam can be saved by the jackets than the uses of the jackets and reheaters call for, to make such practice profitable, especially if the conditions are carefully considered as to ratios of expansion, steam pressures, and the relation between the jacket heat and the working steam. In this connection, certain very high authorities, both in this country and abroad, have concluded that when properly designed and applied, steam jackets will use from 4% to 6% of the total steam sent to the engine in single cylinder condensing engines under the ordinary ratios of expansion, and in compound and in triple engines from 9% to 12%, although a few cases have shown as high a consumption in the jackets as 15% of the total steam supplied to the engine. The pumping engine holding the world's record for economy, September 1, 1905, of 163,925,000 ft. lbs. per 1,000,000 heat units, or of 10.335 lbs. of dry steam per horse power per hour condenses in its jackets 15.45% of the total steam used. Its pressures when the record was taken were as follows:

Steam at throttle, 185 lbs. gauge.
Steam in first receiver, 31 lbs. gauge.
Steam in second receiver, 11 lbs. absolute.
Steam in high pressure jacket, 185 lbs. gauge.
Steam in intermediate jacket, 74 lbs. gauge.
Steam in low pressure jacket, 5 lbs. gauge.

As a result of many records the fact seems to be that single cylinder condensing engines will save from 2 to 3 times net, the

amount of steam used in the jackets; compound engines about twice as much net, the amount of steam used in the jackets; and triples about as much net, as the jackets use.

The conditions of pressure under the above mentioned saving of steam by the use of jackets, is laid down as follows by the British Board of Admiralty:

Type of Engine.	Gauge Pressure.
Single cylinder condensing	90 lbs. initial.
Compound condensing	120
Triple condensing	160
Quadruple condensing	225

Rotative speed does not seem to affect the case if the proper conditions otherwise exist. There are records of compound engines at 250 revolutions per minute, using 6 per cent of the total steam in the jackets and saving 9 per cent net; or, the jackets use 6 per cent of the total steam sent to the engine when jackets are used, and save 15 per cent over what the engines will do without steam jackets, thus making a net economy or profit by the jackets in that case of 9 per cent. The entire desirability of steam jackets seems to be a question of their uselessness or their usefulness, and these qualities are in turn dependent upon their design and arrangement. It is obvious that well jacketed heads expose a generous area of surface in proportion to the rather limited body of the initial steam and will have a more decided effect in furnishing latent heat for entrained water, than merely jacketing the sides which are not exposed until a goodly portion of the stroke has been accomplished; and after all the prevention of initial condensation is the most valuable use of a jacket, for one simple reason among others, that is, if the steam starts dry, the constantly falling pressure due to expansion presents no tendency to moisture which cannot be instantly taken care of under the lowering pressure and the constantly high temperature within the cylinder and jackets.

With a 30 inch cylinder for example, when the piston has moved $2\frac{1}{2}$ inches from the beginning of the stroke, the initial steam will represent about one cubic foot of quantity, while with a jacketed head the heating area of the head will represent about 5 square feet of surface.

With jacketed sides and with no head jackets, the same amount of motion of the piston would give one cubic foot of steam to be heated and a little over $1\frac{1}{2}$ square feet of heating surface. And besides this, with the jacketed head the radiation would have to go $2\frac{1}{2}$ inches only to completely penetrate the steam, while with the side jacket the radiation would have to go 15 inches to reach the entire body of steam. These results would be more or less modified by the admission ports and the heat of the incoming steam, but this would be robbing Peter to pay Paul, and Paul's share would not represent the full robbery of Peter, either.

Some of the apparent inconsistencies of the reported steam consumption in the jackets might possibly be explained by just such differences in the application of steam jackets. Such jackets are of all kinds and qualities, according to the more or less successful attempts of the designer; a high efficiency of jacket from a well jacketed cylinder head, working upon the initial steam at the instant of its admission and near the commencement of the stroke, wherein a comparatively limited area is giving a highly useful effect, may result in the use of a very moderate percentage of the total steam, but counting heavily in the general economy of the engine; while indifferently jacketed heads and a thoroughly jacketed barrel, or a well jacketed barrel and no jackets at the heads, might very easily result in a great deal of jacket condensation with no very great gain in the general economy of the engine. Records show all the way from 1 per cent to 20 per cent of the total steam used in the jackets according to the design of the engine, the design and application of the jackets, and the hundred and one conditions under which the entire outfit might be made to operate.

It must always be borne in mind that the lower the grade of refinement in design in a steam engine the easier it is to improve

its operation. And the higher the degree of refinement the more difficult and also the closer must the conditions be accommodated. With an engine taking steam full stroke, and regulating its operation by raising or lowering the initial pressure, it is already working under about the worst conditions it can, and damage to its economy or an increase in its efficiency is rather difficult to accomplish without some departure from its scope or plan. By applying a cut-off to such an engine, an immediate improvement is observed, but this is accompanied by the fact that the load must be adjusted within certain limits in order that the best results may be obtained. Now, compound this cut-off engine, and a still greater improvement is obtained, but a still closer line must be drawn for its operation and manipulation. Then make it a triple expansion engine, and the lines lie still closer to conditions for satisfactory work. Again, make the engine quadruple expansion, and a still more exacting adaptation is needed for the best work.

With steam jacketing, after the cut-off and expansion has been introduced, the greatest gain is perceptible with the single cylinder cut-off engine, when the range of temperature represented by the initial and terminal pressures is the greatest within one cylinder. And, as the multiple grade of the engine rises, to compound, to triple, and to quadruple, so that for any certain aggregate rate of expansion the range of temperature in any one cylinder becomes less and less, the advantages of steam jacketing gradually decrease, but with the higher ratios of expansion, especially in pumping engines, where the net work and its relation to the indicated horse power can be so closely watched and noted, it has so far at least been deemed necessary for acceptable results.

In making a comparison in the matter of steam jacket value and price of a pumping engine, the following will be instructive in a practical consideration of the subject:

Supposing the contract price of a triple expansion pumping engine of the highest type of efficiency as shown in regular practice, to be, say, $93,000 ready for service. The engine to be of 750 indicated horse power, and to be able to develop a horse

power with the consumption of 10.50 lbs. of steam per hour including jackets, making 7,875 lbs. of steam per hour. Also supposing that the jackets used 10% of the total steam, leaving 7,087.5 lbs. of steam per hour as the quantity required for consumption within the cylinders. Then upon this basis, allowing that without steam jackets, the moderate amount of 20% is taken for condensation within the cylinders, there would be required 8,834 lbs. of steam per hour if jackets were not used, instead of the above mentioned 7,875 lbs. Then at the practical everyday evaporation under 150 lbs. steam pressure of 8 to 1 the difference in coal would be 1.43 tons per day of 24 hours, many of such engines generally running 24 hours per day. At $3 per ton this will capitalize at $31,337 at 5% per annum. This amount deducted from the price of $93,000 for the jacketed engine, will produce $61,663, which is a price very much below the cost and value of an unjacketed engine of otherwise equal quality, design and capacity.

There is a jacket record in a well established case, involving what might be called modern practice, and of very recent years, as low as 9¼% and, as indicating the important effect of steam jacket efficiency in proportion to steam used in the jackets, the following table of steam per indicated horse power per hour and of duties is given. A practical demonstration was made in a recent case bearing directly upon this portion of the subject which clearly illustrates the importance of not using too much steam in the jackets. The duty of a certain pumping engine had not been satisfactory, but a strong belief existed that better results could be shown if the machine was operated under more appropriate conditions. When first examined, the engine was running at normal speed with the throttle valve wide open and the pressures as follows:

Steam at Throttle, Gauge.	First Receiver, Gauge.	Second Receiver, Gauge.	High Pressure Jacket, Gauge.	Intermediate Jacket, Gauge.	Low Pressure Jacket, Gauge.
155	35	4	152	70	27

The first thing done was to ascertain the amount of steam being condensed in the jackets, and at the same time run a short duty test per 1,000 lbs. of steam. The jacket steam was found to be about 18% including receiver coils, which was at once considered to be entirely too high. Then after some careful experimenting, and several tests run under different conditions, the final test for duty was carried out, and the best results of which the engine seemed capable were obtained with the pressures as follows:

Steam at Throttle, Gauge.	First Receiver, Gauge.	Second Receiver, Gauge.	High Pressure Jacket, Gauge.	Intermediate Jacket, Gauge.	Low Pressure Jacket, Gauge.
151	36	4½	85	15	2

The following is the table referred to, showing the effects of an increasing consumption of steam within the jackets brought about presumably by different adaptations of the steam jackets to their work, some more effective than others. The table is based upon one set of conditions only, and are 150 gauge pressure, 33 expansions, 5 lbs. absolute terminal, and the consumption per hour of the working steam of 6,334 lbs. within the cylinders, giving 672 indicated horse power. It must be plain that the lower the amount of jacket steam with which the work can be done, the nearer 100% will be the working steam, and the higher the duty will be per 1,000 lbs. of steam consumed, including jackets and receiver coils.

Percentage of Total Steam Used in the Jackets.	Steam per hour per Indicated Horse Power Including Jackets, etc.	Main Pump Duty per 1,000 lbs. of Steam.
9	10.28	183,503,243
10	10.47	181,484,876
11	10.59	179,510,607
12	10.71	177,737,881
13	10.83	175,531,915
14	10.96	173,380,035
15	11.08	171,429,437
16	11.25	169,230,796

There is one very important point to be recognized in the employment of the steam jacket, and that is, when the cut-off valve closes and the working steam is shut within the cylinder, this working steam is completely beyond reach of direct manipulation, so far as saving and returning to the boiler any part or portion of its heat or substance. It is beyond reach and must go its way for better or for worse. But with a steam jacket, after the steam in the jacket has given up sufficient heat to the cylinder walls, and through these walls to the working charge of steam within, the residue of hot water, or whatever remains, may be returned to the boiler, with the satisfaction that it has only given up whatever has been useful, and the remainder or unused portion is still on hand for future usefulness.

When it began to be comprehended that higher steam pressures and higher ratios of expansion appeared to be the road to higher duty, the way seemed clear. But as usual, new difficulties loomed up; in time to be cleared away in their turn. Among these troubles were higher ranges of temperature within the steam cylinder, although at first not understood; and a very unequal distribution of impulse and working strains from the great difference between the initial and the terminal steam pressures. Active forces in nature seek to produce equilibrium, or, in other words, a balance; and taking advantage of these efforts is just where mankind turns such forces to practical account and usefulness. Water on top of a hill will run down until a general level is reached and rest obtained, and we all know what man does with water going from a higher to a lower level. Different degrees or different forces of heat, so to speak, in different parts of the same substance will endeavor to bring the entire mass to a balanced temperature or uniformity of heat throughout the mass.

Initial or incoming steam within a cylinder, at a gauge pressure of 150 lbs., or 165 absolute pressure, the latter pressure obtained by adding to the gauge pressure, the atmospheric pressure of 15 lbs., has a temperature of 365 degrees by the Fahrenheit thermometer. And if, after the cut-off valve has closed, the

steam should be expanded down to 5 lbs. absolute pressure, or 5 lbs. above a perfect vacuum, equivalent to a little more than 19 inches of vacuum, then the temperature of steam at the lower pressure in the absence of extra heat would be 162 degrees, or 203 degrees below the temperature of the initial or incoming steam at the next stroke of the piston. In the absence of some adequate means for preserving the heat of the cylinder walls, as, for example, a steam jacket, it is not difficult to perceive that a great deal of condensation in the initial steam would take place, and that the actual temperature of the cylinder walls will be somewhere at a point between the two extremes of temperature, much below and very damaging to the initial steam, on account of the high temperature of the incoming steam, and the natural tendency of heat towards striking a balance of temperature in all parts of the same mass.

To illustrate this action of the expansion of steam and the difference between the initial and terminal temperatures within an engine cylinder, reference is made to Fig. 14 which represents a complete or absolute pressure of 165 lbs., equivalent to a gauge pressure of 150 lbs. expanding to 33 times its original volume, and down to a terminal pressure of 5 pounds above a perfect vacuum. Or from 165 absolute down to 5 absolute, and thus indicating a range of temperature of 203 degrees if such conditions could be maintained within a steam cylinder after the closing of the cut-off valve. Now if the temperature of the cylinder walls could follow that of the inclosed steam on the downward grade, and finish with the expansion of the steam at 162 degrees, of course, as already pointed out, the incoming steam at 365 degrees would suffer severely from condensation when entering the cylinder with its walls at 162 degrees, and this would take place at every successive stroke of the piston, thus robbing the operation of expansion of a great deal of its natural advantages. The temperature of the cylinder walls, however, would never get down to 162 nor up to 365, but without outside aid from auxiliary heat would strike a sort of fluctuating balance somewhere between the two temperatures;

rising and falling in the neighborhood of the average temperature at each stroke, and reëvaporating a great deal of the water of condensation towards the finish of the stroke.

Considering this rather extreme illustration of the effects of steam expansion, it would not take a great deal of experimenting to determine just about where the practical and useful limits of high pressure and ratio of expansion were to be found with steam used in an ordinary cylinder without steam jackets or some other means of maintaining heat in the cylinder walls. This

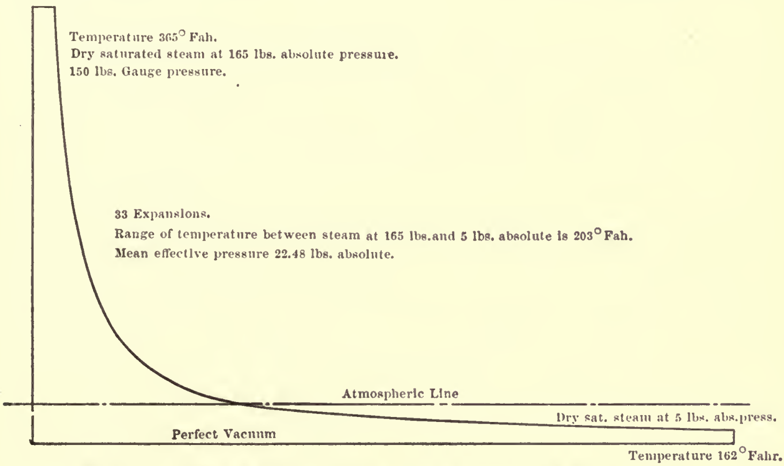


Fig. 14.—The Mariotte Curve showing Range of Temperature.

would probably be with an initial pressure of about 80 lbs. per gauge and with about 4 expansions within any one cylinder. But even with steam jacketing, with the tremendous range of temperature of 203 degrees called for by the differences of pressure for 33 expansions, a good deal of condensation and consequent loss would practically take place in such work, and this drives home the argument that the higher the number of cylinders, the lower the ratio of expansion in one cylinder and the less effective is steam jacketing, while such an attempted range as 203 degrees presents the most profitable field for the steam jacket operation. To more fully illustrate this action, a comparison has been made

in Chapter V in which the Mariotte curve is rather fully explained. This comparison is made between expanding 33 times in one cylinder, and expanding 33 times through three cylinders as exemplified in the present day triple expansion pumping engine. And the comparison for convenience' sake is made with the idea that perfect vacuum is obtained within the single cylinder and within the low pressure cylinder of the triple engine, and it is also assumed in both cases that all of the working steam is accounted for by the diagrams.

An interesting experience by the writer will fit in here in connection with the heat of cylinder walls. A non-condensing, non-jacketed vertical Corliss engine, built by the parent company, was being experimented with for compression; the cylinder was 28 × 60, the engine making 50 revolutions per minute. A second eccentric had been fitted to this engine so as to secure independent adjustment for the exhaust valves and obtain any desired amount of exhaust lead and compression without interfering with the range of cut-off or the operation of the induction valves in any manner. In a word, so that both admission and exhaust valves could be adjusted to the best point of efficiency without interfering with each other. The initial steam shown by the indicator was 85 lbs. per gauge; the point of cut-off was just about 20 inches from the beginning of the stroke; the counter pressure was 3 lbs. per gauge. The indicator was attached, and after some experimenting with the eccentric for the exhaust motion, the rest of the compression adjustments were made by lengthening or shortening the exhaust valve radius rods, the jam nuts being kept loose and the rods turned either way as the demand called, after each indicator card was taken. It was found that 45 lbs. was about the limit of compression without disturbance in the diagram; and when 45 lbs. was reached in the compression curve a small dot appeared where the indicator paused for an instant before the admission line showed that the induction valve was open. By persisting in the attempt at more compression than 45 lbs., a hook, in the line formed where the compression and admission lines joined,

began to appear; and when the exhaust valve was adjusted beyond a certain point of closure, the compression stopped and a slight drop took place which met the admission line, thus forming from the bottom of the card, first a most beautiful compression curve up to 45 lbs. gauge pressure, then a drop hook covering a range of about 4 lbs., and then the admission line took up the work and went up to the initial pressure. Hundreds of experiments were tried and all sorts of changes rung upon the diagrams. The compression curve could be made to glide smoothly into the admission line at just about 45 lbs. pressure; then lengthening the radius rods, the small dot made by the pausing pencil would begin to appear; this dot would grow in size and finally develop into a slight off-set in the diagram; then the hook downwards would begin to appear; and finally the diagram would be repeated many times unchanged as the conditions were established with the exhaust valve closing at an earlier point than the temperature of the cylinder walls would support the compressed steam. This operation could be reversed and the downward hook sent back into a dot; this dot gradually diminishing as the radius rods were adjusted shorter and shorter; until finally upon the reverse experiment the smooth compression curve died out into the admission line as the incoming steam sent the pressure up to the initial point.

The temperatures were as follows:

Initial steam, gauge pressure, 327°.
Counter pressure, 3 lbs. above atmosphere, 220°.
Top of perfect compression curve, 45 lbs. gauge, 292°.
Top of curve with hook, 49 lbs. gauge, 296°.

This seems to indicate, after 45 lbs. was reached, which represents a temperature in the steam of 292 degrees, that the limits of the temperature of the cylinder walls had been also reached; and although the great abundance of work available carried the pressure up 4 lbs. before the formation of compressed steam was stopped and representing a top limit of temperature

of 296 degrees, the temperature of the iron was probably very close to 292 degrees; or between 6% and 7% above the half way point from terminal up to initial.

If this evidence as to cylinder temperature holds substantially correct under most other conditions, then in the examples set forth in Chapter V the initial and cylinder temperatures would be as follows:

High pressure cylinder; no steam jackets.
Initial steam temperature, 365°.
Cylinder temperature, 321°.
Intermediate cylinder; no steam jackets.
Initial steam temperature, 270°.
Cylinder temperature, 238°.
Low pressure cylinder; no steam jackets.
Initial steam temperature, 202°.
Cylinder temperature, 174°.

If these relations of temperature between steam and iron hold good in most cases, it shows how easy it would be to use too much steam in the jackets, and also illustrates the fact that if the jackets are not arranged to get hold of the initial steam instantly and dry it out, they will be extremely likely to do a great deal of useless work.

CHAPTER VII

COAL DUTY OF PUMPING ENGINES

THE coal duty of a pumping engine is a term which, although not strictly correct and scientific, has nevertheless gotten to be such a widely employed expression, that it has come to be used as a technical way of stating the amount of work actually realized in pounds of water raised against the head with the consumption of each 100 lbs. of coal. It is really the duty of the plant; and the economic coal duty actually realized by a pumping plant, expressed in foot pounds of work done per 100 lbs. of coal burned, is influenced by several items apart and aside from the qualifications of the pumping engine doing the work. Prominent among these items are: the efficiency of the boilers or the ability of the boilers to use the heat given out by the burning coal; the actual amount of heat possible to obtain from the coal by its combustion, involving the number of heat units possible to obtain from any certain kind of coal; the skill of the firemen, or the efficiency of the mechanical stokers; the draught in the chimney and the proper manipulation of the damper; the cleanliness and good order of the plant generally.

As boilers go, 72% efficiency represents very good work. That is to say: whatever the utmost number of heat units it may be possible to obtain by analysis of the coal, or, whatever the utmost number of heat units may be, which the coal is capable of giving out under complete combustion, if the boiler can produce under the actual working conditions, between the temperatures of the feed water and the working steam, a quantity of steam which by its contained heat will represent 72% of the total heat of the combustion of the coal burned, then such boilers are doing pretty good work. There are records as high as 81% efficiency; and

with the assistance of economizers in the back flue, to give waste heat to the feed water before it enters the boiler, even so high as 86% efficiency has been recorded. But the higher efficiencies as a rule represent a much higher cost of boiler than a 72% efficiency boiler, and this brings into the account the balance between the interest on the extra cost of boiler and the value of the coal saved by the higher efficiency. It is the old argument in another form between the high and low duty engines at a higher or a lower first cost. That plant will pump water under its own circumstances, for the least cost in money per million gallons sent up the hill, which is the best adapted to its surrounding conditions, including the cost of coal; and the real office of the expert is to make such adaptations before the plant is built, if he gets a chance, rather than to endeavor to make impossible ends meet after the plant is built wrong.

The bearing of the practical boiler efficiency, and the heat units of the coal, may be seen more clearly by the aid of a few figures from actual cases; and men who run pumping engines, who fire boilers, and who make and sell boilers, can readily perceive where the limits of what can really be done are situated.

First Case:

Boiler efficiency 72% of the heat of the coal.

Temperature of the feed, 150°.

Working pressure per gauge, 150 lbs.

Units of heat per pound of coal, 14,360.

Then the boiler would turn into steam 72% of 14,360, which means that there are 10,339.2 heat units available for steam per pound of coal burned, and of course this 72% efficiency includes the furnace as well as the boiler.

The range of heat units between 150° feed and 150 lbs. steam pressure is as follows:

150° feed; heat units per pound above 32°	119.3
150 lbs. steam pressure, gauge above 32°	1,193.5

Then the heat units given by the boiler to each pound of steam

produced, that is, one pound of water evaporated into steam, is 1,074.2, or in even numbers say, 1,074 heat units to be given up to make each pound weight of steam with feed at 150° and into steam at 150 lbs. pressure per gauge.

Then the dry steam possible to be produced per pound of coal under the above mentioned conditions would be at 72% efficiency of boiler and furnace, 9.63 lbs.

And this shows that between 96% and 97% of the duty per 1,000 lbs. of steam would be realized as coal duty of the pumping engine. Or, if the duty happened to be, say, 140,000,000 ft. lbs. per 1,000 lbs. of steam used by the engine, then the actual coal duty would be 96% of the steam duty, or 134,400,000 ft. lbs. per 100 lbs. of coal, which, although something of a margin below the steam duty, would be an exact equivalent under the circumstances.

Second Case:

Boiler efficiency 68% of the heat of the coal.

Temperature of the feed, 150°.

Working pressure per gauge, 150 lbs.

Units of heat per pound of coal, 13,500.

Then the boiler would turn into steam 68% of 13,500, which means that there are 9,180 heat units available for steam per pound of coal burned, and of course this 68% efficiency includes the furnace as well as the boiler.

The range of heat units between 150° feed and 150 lbs. steam pressure is as follows:

150° feed; heat units per pound above 32° 119.3
150 lbs. steam pressure, gauge above 32° 1,193.5

Then the heat units given by the boiler to each pound of steam produced, that is, one pound of water evaporated into steam, is 1,074.2, or in even numbers say, 1,074 heat units to be given up to make each pound weight of steam with feed at 150° and into steam at 150 lbs. pressure per gauge.

Then the dry steam possible to be produced per pound of coal

under the above mentioned conditions would be at 68% efficiency of boiler and furnace, 8.55 lbs.

And this shows that 85.5% of the duty per 1,000 lbs. of steam would be realized as coal duty. Or, if the duty happened to be 140,000,000 ft. lbs. per 1,000 lbs. of steam, used by the engine, then the actual coal duty of the plant would be 119,700,000 ft. lbs. per 100 lbs. of coal, which looks to be very much less than the steam duty, but which is an exact equivalent under the circumstances.

Therefore it will be seen that every change in the number of heat units possible to derive from combustion of the different values of coal, and every change in the practical efficiency of the boilers in every-day operation, in the hands of the men who operate them, marks a difference in the observed coal duty of the plant; and although these differences could be tabulated so as to cover a considerable range of coal and boilers, it would be a very laborious task, and a great deal of valuable space would be filled. It is much more convenient to carry each case by itself from the factors actually obtained in work daily accomplished; but to show how rapidly and completely the changes would take place in coal duty under various conditions of operation, the accompanying table is given in which the steam pressure is taken at 150 lbs. per gauge, the temperature of feed is taken at three different points, the quality of coal as to heat units is taken at four points, and the efficiency of the boilers is taken at 16 different percentages.

With the same duty per 1,000 lbs. of steam given by the engine, 140,000,000 as above, there is a range in coal duty, or duty of the plant, from 105,000,000 to 152,600,000 per 100 lbs. of coal, which is the basis generally reported by the water works superintendent, and this basis seems to be the most rational one because it deals directly with the fuel which is bought and the water which is sold. Besides this the coal-weighing is a natural and convenient operation when the fuel is brought into the fire room for use at the boilers, giving a check against the weights billed by the dealer as time goes on; and the different

gauges and appliances which go with pumping engines afford an easy way of keeping account of the work done by the machinery. The following table shows some of the changes which take place in coal duty according to the fuel and boilers used:

Table showing what percentages of the duty per 1,000 pounds of steam it is possible to obtain with different grades of coal, with different temperatures of feed, and with different efficiencies of the furnaces and boilers.

150 *pounds steam pressure per gauge.*

Heat Efficiency of the Furnaces and Boilers Given in per cent.	Heat Units per Pound of Coal 13,000.			Heat Units per Pound of Coal 13,500.			Heat Units per Pound of Coal 14,000.			Heat Units per Pound of Coal 14,500.		
	Percentages of the Duty per 1,000 Lbs. of Steam at Given Temperature of Feed.			Percentages of the Duty per 1,000 Lbs. of Steam at Given Temperature of Feed.			Percentages of the Duty per 1,000 Lbs. of Steam at Given Temperature of Feed.			Percentages of the Duty per 1,000 Lbs. of Steam at Given Temperature of Feed.		
	100°	150°	175°	100°	150°	175°	100°	150°	175°	100°	150°	175°
65	75	79	80	78	82	84	81	85	87	84	88	90
66	76	80	82	79	83	85	82	86	88	85	89	91
67	77	81	83	80	85	86	83	87	89	86	91	93
68	78	82	84	82	86	88	85	89	91	87	92	94
69	79	83	85	83	87	89	86	90	92	88	93	95
70	80	85	86	84	88	90	87	91	93	90	94	96
71	82	86	88	85	90	91	88	92	95	91	96	98
72	83	87	89	86	91	93	90	94	96	93	97	99
73	84	88	90	88	92	94	91	95	98	94	99	101
74	85	89	92	89	93	95	92	96	99	95	100	102
75	87	90	93	90	95	96	93	98	100	96	101	103
76	88	92	94	91	96	98	95	99	101	98	103	105
77	89	93	95	92	97	99	96	100	102	99	104	106
78	90	94	97	93	98	100	97	101	103	100	105	107
79	91	96	98	95	100	101	99	102	105	101	106	108
80	92	97	99	96	101	103	100	104	106	102	107	109

This table is calculated by taking a range of boiler efficiencies from 65% to 80%, as these percentages cover ordinary, fair, good, and excellent boiler performance. There are percentages below 65, but when there are, it indicates as a general thing that something is seriously wrong with the boiler plant and needs looking after. There are percentages above 80, but such a condition of

affairs indicates that an unusually high priced boiler plant is in use, or unusually high priced fuel is being burned; both of which ought to be properly located and adapted, or they will not pay to operate.

These percentages of efficiency, found in the first column at the left hand side of the table, represent the percentage in different cases in good practice of the total heat of combustion possible to obtain from the coal, which is realized and used by the furnace and the boiler heating surfaces in producing the steam. The temperature of the feed water going into the boiler, and the number of heat units, known as British thermal units, contained in one pound weight of the water to produce the temperature given in the table, is the starting point at which the heat from the coal begins to operate. And the number of heat units necessary to produce one pound weight of steam at the pressure given at the top of the table, 150 lbs. per gauge, is the finishing point reached by the heat from the coal in producing the steam which comes out of the boiler.

The difference between the number of heat units in the feed water going into the boiler, and the number of heat units contained in the steam going out of the boiler, shows how much heat must be furnished by the burning coal to make the steam; and this is reduced for convenience in making the table to a unit of one pound in weight.

The temperature of the feed water given in the table at three different points, viz: 100°, 150°, and 175°, are probably as good as will be generally found in practice, but the operations which establish the table may be applied to any other temperatures desired. It is obvious that the higher the temperature of the feed, the less will be the number of heat units required per pound of steam from the fuel; and the scale of operation of these three different temperatures appears as follows:

One pound of the feed water —

At 100° temperature, contains 68.08 heat units above 32°.
At 150° temperature, contains 118.31 heat units above 32°.
At 175° temperature, contains 143.50 heat units above 32°.

One pound weight of the steam, gauge pressure 150 lbs., contains 1,193.4 heat units above 32°.

Then the difference in heat units required to make one pound weight of steam from the feed water at the temperatures given in the table, into steam at 150 lbs. gauge pressure, would be,

With feed at 100° — 1,125.32 heat units per pound.
With feed at 150° — 1,075.09 heat units per pound.
With feed at 175° — 1,049.90 heat units per pound.

Having determined how much heat will be necessary to make one pound weight of steam at 150 lbs. gauge pressure, with different temperatures of feed, the next step is to determine how much heat may be derived from the combustion of the coal. And for convenience in making a practicable table, four heat values of coal are taken, although the operations forming the table can be applied to any heat value of coal. The heat values taken are at, 13,000 heat units per pound of coal; at 13,500 heat units per pound; at 14,000 heat units per pound; at 14,500 heat units per pound; which is to say, that in the table, four grades of coal are considered which upon complete combustion will develop the number of British thermal units per pound set forth in the table in each case. It being understood, as already pointed out, that the determined, or theoretical power of the coal is very much above the possible results to be found in actual operation with furnaces and boilers, on account of the various losses, such as imperfect combustion, more or less, of the gases given off by the burning fuel; the actual loss of some of the coal by its being unburned and therefore remaining among the ashes and refuse from the furnace; there are also losses from imperfect transmission of heat through the material of the heating surfaces of the boiler to the contained water, even where the combustion, and the production of heat, are nearly perfect. And it is the general result found in practice which marks the difference between the real amount of heat of perfect combustion and the amount of heat actually present in the steam produced, which go to make up the column of percentages at the left hand side of the table,

showing the practical efficiency of the boiler and furnace as a steam-making apparatus, in proportion to the heat value of the coal burned upon the grates. Any boiler efficiency known or selected from the table in per cent in the left hand column, will apply to any of the heat values of the coal given, or any other heat value will show how many heat units per pound of coal the boiler is having the use of in producing steam. Then this net number of heat units derived from a pound of coal by the boiler, divided by the range of heat units between the feed and the steam pressure, will indicate the actual evaporation of pounds of water per pound of coal under the working conditions; and the decimal point in the rate of evaporation so found, if moved one place to the left will express the percentage of the duty per 1,000 lbs. of steam which will be obtained per 100 lbs. of coal.

If the decimal point be placed between the two figures where there are two figures only, or at the left side of the right hand figure where there are three figures, found in this table expressing percentages of duty in the columns under temperatures of feed, the result will show the evaporation per pound of coal going on in the boiler, with the heat value of the coal and the efficiency of boiler given above, and opposite, the percentage of duty selected.

Where the percentage of the duty per 1,000 lbs. of steam reaches in the table to 100 or above, and which it does in a number of places, this indicates that the actual evaporation in the boiler, either from high efficiency of boiler or from high heat value of the coal, or both, is more than 10 to 1, the commonly accepted basis for making comparisons in duty tests of pumping engines.

Aside from the heat value of the coal burned in a water works plant, there is also a money value. And the question is often brought up in considering the practical value of economic duty, especially where it involves a comparatively high cost of machinery, as to what price per ton as coal is commercially bought, it will pay to go in purchasing supplies. Of course the better

the quality the more it is worth, but there is a dividing line, where the heat units possible to obtain from the combustion balances against the price in money which must be paid for any particular grade of fuel. And as heat units are eventually obtained for the money paid out, a comparison between these two items will be in order; but the question is too complex to go into extensively within the limits of this book, although a meager outline will be given to indicate the direction possible to follow where the necessary information is available.

Bituminous Coal

With a bituminous coal showing say 14,500 heat units per pound and costing say $3 per net ton of 2,000 pounds, the buyer will receive 96,666 heat units for a cent. And taking this as a basis of value for other coals, the following table of price or money value for different grades of coal in proportion to the ability to furnish heat appears:

Heat Units per Pound on Complete Combustion.	Price per Net Ton of 2,000 Pounds Delivered in the Boiler Room.
14,500	$3.00
14,250	2.94
14,000	2.89
13,750	2.84
13,500	2.79
13,250	2.74
13,000	2.68
12,750	2.63
12,500	2.58
12,250	2.53
12,000	2.48

These results are obtained by multiplying the heat units from one pound of the coal at the price of $3 per ton, which is 14,500 and is taken as a standard, by 2,000, the number of pounds in a net ton; and this result divided by 300 cents ($3) gives the number of heat units for one cent based upon a fair market price for a

good quality of soft coal, and amounts to 96,666 heat units for a cent. Then the heat units for a cent, in this particular case 96,666, divided into the heat units in one ton, which, 2,000, multiplied by the number of heat units per pound of the coal, will give the price value of the coal in cents, which pointed off in two places will give dollars and cents per ton, upon the basis of $3 for the higher quality. If the price of the higher quality varies in different markets or places then the comparative price for the other grades will also vary, but the proportion of price to heat units will hold good whatever the basic price or the location of the market where the coal is bought.

The results would be modified to a slight extent by the difference in coal arising from other elements than the mere difference in total heat units; as, for example, fixed carbon in the coal, which is the most valuable portion of the fuel from a practical standpoint because it is the easiest to accommodate to the combustion within the furnace; whereas the gaseous portion of the coal although equally valuable under proper conditions is more difficult to manage economically. In soft coal the fixed carbon or solid portion of the fuel varies from 54% to 83% while the gaseous or volatile matter varies from 12% to 48%, and hence the same arrangement for furnace, air, and draught will not answer for all grades of coal. But with fixed carbon varying from 68% to 78% and a low amount of sulphur, very good results will be found according to the table, and based upon the total amount of heat units per pound. It is not difficult to ascertain the difference in quality of coals, and it will no doubt pay to give some attention to the subject when making contracts for supplies of fuel for water works purposes.

Anthracite Coal

With an anthracite coal showing say 14,000 heat units per pound on total combustion, and costing say $4.50 per net ton of 2,000 pounds, the buyer will receive 62,222 heat units for a cent. And taking this as a basis of value for other grades, the following

table of price or money value for different grades of coal in proportion to the ability to furnish heat appears:

Heat Units per Pound on Complete Combustion.	Price per Net Ton of 2,000 Pounds Delivered in the Boiler Room.
14,000	$4.50
13,750	4.42
13,500	4.36
13,250	4.26
13,000	4.17
12,750	4.10
12,500	4.02
12,250	3.94
12,000	3.86

This table is constructed by the same method as the bituminous table immediately preceding, and subject to the same restrictions regarding elements in the coal, but the difference against the anthracite in price is not quite so great as apparent at first glance, for the reason that the fixed carbon is about 20% higher in anthracite than in bituminous on the average, so where freight rates, losses from storage and from weather, and other factors play a part, sometimes the higher priced anthracite is the most economical in proportion to water pumped. As an example of this two records are given in actual practice based upon the generally accepted factor of coal consumed per indicated horse power per hour, and which were as follows:

In one case there was consumed 1.02 pounds of coal per indicated horse power, and in the other case 1.98 pounds of coal per indicated horse power per hour. Both records were obtained as nearly as could be under similar conditions in actual water works pumping. The fuel calling for 1.98 pounds was anthracite slack; and the fuel calling for 1.02 pounds was a good quality of anthracite coal. The slack was $1.50 per net ton, and the regular coal was $4.50 per net ton. The analysis of the fuels for heat units on complete combustion was as follows:

Heat units for the slack, 11,000 per pound.

Heat units for the coal, 14,000 per pound.

(These are closely approximate round numbers.)

In the case of the slack the buyer obtained 146,000 heat units for a cent; and in the case of the coal the buyer obtained 62,000 heat units for a cent.

The plant using the slack consumed 363 heat units per horse power per minute on the coal duty basis, and the plant using the regular anthracite coal consumed 238 heat units per horse power per minute upon the coal duty basis. The efficiency of the boilers was 70% in the case of the slack, and 80% efficiency in the case of the coal. This gives 8,579,970 ft. lbs. of work for a cent with the slack, and 13,200,000 ft. lbs. of work for a cent with the regular coal.

The horse power was about the same, and has been equalized in the calculation at 625 indicated horse power, which makes the slack cost 95 cents per hour for the power required, and the coal cost $1.63 per hour for the same power. But the slack is used practically at the point of production while the coal is freighted several hundred miles, and if the price of the slack should be raised $1.25 per ton, making it cost $2.75 per ton, then the cost of the power would be $1.70 per hour as against $1.63 per hour for the coal.

Or to put it another way, it will require 27% more of the slack to obtain the same number of heat units that a pound of the coal is good for, and even at $1.50 per ton, the increase in the quantity would carry the price of the slack up to an equivalent of $1.90 per ton, and the increased quantity would allow a freight rate of only 65 cents per ton, to keep the cost for the power per hour down to $1.70 or 7 cents above the cost of the coal upon the same basis. It is extremely doubtful if the slack can be freighted the distance required for 65 cents per ton, and aside from this there would be an increased quantity of ashes to be taken care of and to dispose of in the operation of the plant.

It is sometimes convenient to know at a glance what the duty will be for any certain rate of actual evaporation in the boilers, under every-day working conditions usually found in the boiler room of a pumping plant. In large engines and under good

boiler conditions, yearly duties are maintained as high as from 120,000,000 to 135,000,000 ft. lbs. per 100 lbs. of coal burned; and it must be borne in mind that no matter how high a steam duty a pumping engine may show, the results at the boilers in producing the steam have a strong controlling influence upon the yearly reports of economical operation. To illustrate the difference that can exist between steam duty per 1,000 lbs. and coal duty per 100 lbs. based upon the actual weight of fuel required, the accompanying table is given.

Duty on coal burned in the furnaces of the boilers, for different rates of evaporation, and for the following duties per 1,000 pounds of steam.

Duty in Foot Pounds per 1,000 Pounds of Dry Steam.	Duty in Foot Pounds per 100 Pounds of Coal for Different Rates of Actual Evaporation in the Boilers.			
	8 Pounds.	8.5 Pounds.	9 Pounds.	9.5 Pounds.
40,000,000	32,000,000	34,000,000	36,000,000	38,000,000
50,000,000	40,000,000	42,500,000	45,000,000	47,500,000
60,000,000	48,000,000	51,000,000	54,000,000	57,000,000
70,000,000	56,000,000	59,500,000	63,000,000	66,500,000
80,000,000	64,000,000	68,000,000	72,000,000	76,000,000
90,000,000	72,000,000	76,500,000	81,000,000	85,500,000
100,000,000	80,000,000	85,000,000	90,000,000	95,000,000
110,000,000	88,000,000	93,500,000	99,000,000	104,500,000
115,000,000	92,000,000	97,750,000	103,500,000	109,250,000
120,000,000	96,000,000	102,000,000	108,000,000	114,000,000
125,000,000	100,000,000	106,250,000	112,500,000	118,750,000
130,000,000	104,000,000	110,500,000	117,000,000	123,500,000
135,000,000	108,000,000	114,750,000	121,500,000	128,250,000
140,000,000	112,000,000	119,000,000	126,000,000	133,000,000
145,000,000	116,000,000	123,250,000	130,500,000	137,750,000
150,000,000	120,000,000	127,500,000	135,000,000	141,500,000
155,000,000	124,000,000	132,750,000	139,500,000	147,250,000
160,000,000	128,000,000	136,000,000	144,000,000	152,000,000
165,000,000	132,000,000	140,250,000	148,500,000	156,750,000
170,000,000	136,000,000	144,500,000	153,000,000	161,500,000
175,000,000	140,000,000	148,750,000	157,500,000	166,250,000
180,000,000	144,000,000	153,000,000	162,000,000	171,000,000
185,000,000	148,000,000	157,250,000	166,500,000	175,750,000
190,000,000	152,000,000	161,500,000	171,000,000	180,500,000
195,000,000	156,000,000	165,750,000	175,000,000	185,250,000
200,000,000	160,000,000	170,000,000	180,000,000	190,000,000

CHAPTER VIII

ACTUAL CONDITIONS OF PUMPING

WITH a properly designed and well built machine, its actual steam economy should be maintained at very nearly the maximum, and no doubt it is in a great majority of cases; in fact, unless cutting or other damage to valves and pistons takes place, the initial efficiency cannot be lowered very much without gross inattention, or without a very serious departure from the steam and water pressures for which the engine was built. But where, as in the usual annual report of the water works superintendent, the statement of duty is in terms of coal, it will be seen at once that there are a good many chances for losses of various kinds. When operated under good conditions there is not very much difference in boilers so far as efficiency is concerned, but when boilers are worked under bad conditions there is a different story to tell. There are three prominent items affecting the actual coal duty of a pumping plant pertaining to the boilers. They are, overworked boilers; underworked boilers; and various kinds of coal. And for any shortcomings in any of these or other directions belonging to the boilers, the engine is not responsible.

Another reason why pumping engines sometimes fall short in duty in regular service is that they are not always properly proportioned to the work to be done. This cannot be helped sometimes. The future must sometimes be reckoned for, and when an engine is put in, it must sometimes be larger than present needs demand. But the contractor is entitled to test his engine under the best conditions for which it is built, and therefore when the experts get away and the machine is put into the regular service the ideal conditions are destroyed and

unfavorable conditions substituted. Chicago, for example, had such an experience some years ago. The engines advertised for were proportioned, according to the specifications, to pump against a head 50% greater than really developed in regular service, with the result that triple expansion engines were placed under conditions where compound engines with smaller steam ends would have undoubtedly done much better economic work. What happened, apparently, was that the high and intermediate pressure cylinders did so much of the work that there was only a low temperature fog left for the low pressure cylinder to handle, and the third plunger was largely operated through the medium of the crank and connecting rod, dragging the low pressure piston along incidentally. No comments are offered upon the facts; the reference is only used as an illustration of how disappointment may be met with, even though a high type of machine is secured. It further illustrates how a duty test run by experts under proper conditions may be greatly discounted in every-day operation, and where "somebody blundered," but where the engines and the experts were not to blame for the shortcomings.

Reference has just been made to overworked and underworked boilers; and this suggests an appropriate illustration upon this point as to how much the required amount of heating surface in boilers will vary to suit different rates of economy in the steam consumption of the pumping machinery, covering the different types and classes of pumping engines as follows:

Compound non-condensing, direct acting.
Compound condensing, direct acting.
Low duty triple expansion, direct acting.
High duty compound condensing, direct acting.
High duty triple expansion condensing, direct acting.
Cross-compound condensing, crank and fly wheel.
Double compound condensing, crank and fly wheel.
Triple expansion condensing, crank and fly wheel.
Quadruple expansion condensing, crank and fly wheel.

The relation between the difference in economical duty of the various types and classes of pumping engines, and the amount of boiler required, may be conveniently shown by the accompanying table; the measurable amount of boiler needed being positively indicated by taking 10 square feet of heating surface as ordinarily measured for each boiler horse power. The table is based upon the fact repeatedly demonstrated by easily evaporating a little over 10,500 lbs. of water per hour in a pair of water tube boilers having 3,500 square feet of heating surface, that in good ordinary practice, one square foot of heating surface will evaporate 3 lbs. of water per hour, from 150° temperature of feed into steam at 150 lbs. gauge pressure. And therefore 10 square feet will evaporate 30 lbs. of water per hour as above, and this amount of evaporation is taken as one boiler horse power. This at least is a good basis, safe in most cases; but if at any time caution, or any special reasons, should suggest an increase in heating surface, any certain desired percentage increase can be readily added without disturbing the relations of the different rates of economy; as, for example, if it should be decided that 2½ lbs. of water per square foot of heating surface per hour is all that it would be safe to reckon upon, then 20% added to the boiler horse power of the table would provide for such a case. Or, if, to go to extremes somewhat, it was thought that 2 lbs. of water per hour per square foot of heating surface was the limit, then 50% added to the table would meet the demands for boiler capacity. But the writer believes that with properly constructed and arranged boilers, the table will answer all reasonable purposes.

The economy of the pumping engines, which is entirely independent of the working of the boilers, is expressed in foot pounds duty per 1,000 lbs. of steam, in the left hand column of the table.

The second column of the table is calculated by ascertaining the steam per pump horse power per hour by dividing the duty given in the table by 1,000, which gives the duty per pound of

steam consumed; and then dividing the foot pounds of work of one horse power per hour, equal to 60 times 33,000 or 1,980,000 foot pounds, by the duty per pound of steam consumed already ascertained. The result will be the steam per pump horse power per hour at the rate of duty selected from the table, or any other duty may be treated in the same manner.

Then the steam per pump horse power per hour divided by 30, which is the steam per boiler horse power per hour, will give the required boiler horse power, per pump horse power of the pumping engine.

Boiler Horse Power required for each Pump Horse Power; counting 10 square feet of Heating Surface for each Boiler Horse Power, for the following Duties of Engines.

Duty in Foot Pounds per 1,000 Pounds of Steam.	Pounds of Steam per Hour per Pump Horse Power.	Boiler Horse Power per Pump Horse Power.
40,000,000	49.50	1.63
50,000,000	39.60	1.32
60,000,000	33.00	1.10
70,000,000	28.38	0.94
80,000,000	24.75	0.83
90,000,000	22.00	0.74
100,000,000	19.80	0.66
110,000,000	18.00	0.60
115,000,000	17.21	0.57
120,000,000	16.50	0.55
125,000,000	15.80	0.52
130.000,000	15.20	0.51
135,000,000	14.66	0.49
140,000,000	14.14	0.47
145,000,000	13.65	0.46
150,000,000	13.20	0.44
155,000,000	12.77	0.43
160,000,000	12.37	0.41
165,000,000	12.00	0.40
170,000,000	11.65	0.39
175,000,000	11.31	0.38
180,000,000	11.00	0.37
185,000,000	10.70	0.36
190,000,000	10.42	0.35
195,000,000	10.01	0.34
200,000,000	9.90	0.33

There are two phases of the question of the power required for pumping water which are closely relative to the foregoing

table, and these are, the pump horse power for various quantities of water pumped against different pressures, and the indicated steam power developed in the steam cylinders corresponding to the quantities and pressures given for the pump end of the machine; and at different mechanical efficiencies of the engine as a whole. The powers given for the water ends of the engines are the net powers represented by the weight of the quantity of water per minute, multiplied by the total head in feet including suction and friction in the water, as shown by the pressure gauge, or, in other words, the actual power developed by the water end of the working engine, and exclusive of the friction of the machine.

The table herewith, of water end or pump horse power, is based upon an easily remembered rule which may be quite readily committed to memory, and is based upon the fact that with 2,500,000 gallons of water per 24 hours the actual horse power of the water column is one horse power for each pound of water pressure, reckoning the total load and including the suction lift. This total load is made up by accurately reading the water pressure gauge, and then adding to the reading in pounds, the distance from the center of this gauge down to the level of the water in the pump well, this latter distance converted into lbs. pressure by dividing the vertical distance in feet by 2.31 feet per pound pressure. This will give a result within one per cent of what an accurate calculation made in the usual way will produce; and when it is considered how extremely difficult it is to establish absolutely accurate pressures from reading gauges or observing mercury columns, with all the necessary corrections for mercury at different temperatures, the eccentricities of Bourdon gauge springs, and other chances for error so well known to the expert (the more expert he is, the more errors he knows about), a margin of one per cent, and that against the machine, may be considered pretty close. Then whatever multiple the quantity of water to be considered, is of 2,500,000, so will the horse power figures be determined.

Table Showing Horse Power of Water End of Pumping Engines.

Capacity in U. S. Gallons per 24 Hours.	Total Water Load Against Plungers in Pounds Pressure, Including suction.											
	40	50	60	70	80	90	100	110	120	130	140	150
	Horse Powers of the Water Ends.											
1,000,000	16	20	24	28	32	36	40	44	48	52	56	60
1,500,000	24	30	36	42	48	54	60	66	72	78	84	90
2,000,000	32	40	48	56	64	72	80	88	96	104	112	120
2,500,000	40	50	60	70	80	90	100	110	120	130	140	150
3,000,000	48	60	72	84	96	108	120	132	144	156	168	180
4,000,000	64	80	96	112	128	144	160	176	192	208	224	240
5,000,000	80	100	120	140	160	180	200	220	240	260	280	300
6,000,000	96	120	144	168	192	216	240	264	288	312	336	360
7,000,000	112	140	168	196	224	252	280	308	336	364	392	420
8,000,000	128	160	192	224	256	288	320	352	384	416	448	480
9,000,000	144	180	216	252	288	324	360	396	432	468	504	540
10,000,000	160	200	240	280	320	360	400	440	480	520	560	600
11,000,000	176	220	264	308	352	396	440	484	528	572	616	660
12,000,000	192	240	288	336	384	432	480	528	576	624	672	720
13,000,000	208	260	312	364	416	468	520	572	624	670	728	780
14,000,000	224	280	336	392	448	504	560	616	672	728	784	840
15,000,000	240	300	360	420	480	540	600	660	720	780	840	900
16,000,000	256	320	384	448	512	576	640	704	768	832	896	960
17,000,000	272	340	408	476	544	612	680	748	816	884	952	1020
18,000,000	288	360	432	504	576	648	720	792	864	936	1008	1080
20,000,000	320	400	480	560	640	720	800	880	960	1040	1120	1200
22,000,000	352	440	528	616	704	792	880	968	1056	1144	1232	1320
25,000,000	400	500	600	700	800	900	1000	1100	1200	1300	1400	1500
30,000,000	480	600	720	840	960	1080	1200	1320	1440	1560	1680	1800
35,000,000	560	700	840	980	1120	1260	1400	1540	1680	1820	1960	2100
40,000,000	640	800	960	1120	1280	1440	1600	1760	1920	2080	2240	2400

As an example from the foregoing table, 5,000,000 is twice 2,500,000, and so the horse power for the 5,000,000 will be double the water load in pounds; and for 10,000,000 gallons will be 4 times the water load in pounds. This means that the quantity of water in U. S. gallons per 24 hours for which the horse power is wanted, is to be divided by 2,500,000 and the result multiplied by the water load in pounds pressure, the product giving the horse power of the water column. After a little practice this rule may be readily used mentally, and the power determined with great quickness and accuracy.

If it is desired in using this table, to ascertain the horse power for any quantity of water given in the left hand column of the table but for a water pressure load not given in the table, then find the horse power opposite the quantity of water considered, and under 100 lbs. pressure; point off two places of decimals from the right and multiply by the pressure given or desired; the result will be the horse power.

The power of the steam end of the pumping engine, called the Indicated Horse Power, is one of the important guides in considering the economical efficiency of the machine; and the pounds of steam per indicated horse power per hour as showing the power produced in proportion to the steam supplied to the engine is closely observed. The relation between the indicated horse power and the pump or water end horse power, discloses the mechanical efficiency of the machine, or, in other words, shows how much of the power produced in the engine by the heat energy is utilized as useful work. This difference between what is shown to exist as power in the steam cylinders, and what is shown to exist as useful work in the pump cylinders, is lost in friction either of the working parts of the machine itself, or of the water passing through the pumps, or both. And many disappointments have been met with after developing a very nearly perfect steam performance, by seeing the heat efficiency of the steam engine so dearly bought, disappear in the doing of useless work within the machine itself. This mechanical efficiency of pumping engines, of capacities ranging from 6,000,000 U. S. gallons to 35,000,000 U. S. gallons per 24 hours, varies all the way from 88.6% to 96.8% in machinery designed and built by the best and leading engineers and establishments in this country.

The records plainly show the close relation between the mechanical and steam efficiencies, for with the best and the largest engines, the low mechanical efficiency and the lower ranges of duty go hand in hand; when the former is in the neighborhood of 88% the latter is around 157,000,000 ft. lbs., and the high duty record up to April, 1906, was held slightly

above 179,000,000 ft. lbs. with a mechanical efficiency of 96.8% in the machine. The size of the machine does not always control, because the record shows a 6,000,000 gallon engine with a mechanical efficiency of 93% and a 35,000,000 gallon engine with 88% as its efficiency record. Nor does the water load or pressure altogether govern in the matter, for the record shows 252 lbs. water load with 93% and 54 lbs. pressure with 96.5% mechanical efficiency. The probabilities are that the larger machine under the same working load will show the greater efficiency, and the smaller machine under any load, other things being equal, will show the lesser efficiency.

The mechanical efficiency of pumping engines, or their net effectiveness for doing useful work, shown in the extreme right hand column of the accompanying table of indicated horse power, is the percentage which the water end horse power is of the steam end horse power; and may also be expressed by an ordinary fraction in which the water end horse power is the upper term and the steam end horse power is the lower term, or, the numerator and the denominator; as, for example, with a 10,000,000 gallon engine working against 100 lbs. pressure, the water end horse power is 400 and the steam end horse power is 450 at 89 per cent mechanical efficiency. Then the ordinary fraction would be $\frac{400}{450}$ or by placing a decimal point after the 400 and annexing ciphers to the numerator, divide by the denominator in the usual way for transposing the ordinary fractions into decimal fractions, and the mechanical efficiency is expressed in per cent, in this case resulting in 89 per cent.

The table showing mechanical efficiency, indicated horse power, water load against the plungers, and daily capacity follows:

Table of Mechanical Efficiency of Pumping Engines, Showing the Indicated Horse Power in the Steam Cylinders.

Capacity U. S. Gallons per 24 Hours.	Total Water Load Against the Plungers in Pounds Pressure, Including Suction.												Mech. Eff. in % of Indicated Power.
	40	50	60	70	80	90	100	110	120	130	140	150	
	Indicated Horse Power of the Steam Cylinders.												
1,000,000	20	25	30	35	40	45	50	55	60	65	70	75	80
1,500,000	30	37	44	52	58	66	74	81	90	96	104	111	81
2,000,000	40	50	60	70	80	90	100	110	120	130	140	150	82
2,500,000	48	60	72	82	96	110	120	133	145	160	170	180	83
3,000,000	55	71	84	100	114	128	143	157	171	185	200	214	84
4,000,000	75	94	113	132	150	170	188	207	226	245	264	283	85
5,000,000	93	116	140	161	186	210	234	255	280	304	325	350	86
6,000,000	110	138	166	205	220	248	276	303	330	360	386	415	87
7,000,000	129	161	194	225	257	290	322	355	387	418	450	484	87
8,000,000	146	182	218	255	291	328	364	400	435	474	510	547	88
9,000,000	164	205	245	286	328	368	410	450	492	532	572	614	88
10,000,000	146	225	270	315	360	405	450	495	540	585	630	675	89
11,000,000	198	247	296	346	395	446	495	543	592	642	690	740	89
12,000,000	214	266	320	375	426	480	534	585	640	692	750	800	90
13,000,000	231	289	346	404	464	520	579	638	694	745	809	869	90
14,000,000	246	308	370	430	494	554	617	676	740	800	860	923	91
15,000,000	264	330	396	461	529	593	660	726	790	860	922	990	91
16,000,000	278	348	418	488	556	627	697	767	832	906	975	1043	92
17,000,000	296	370	444	520	591	666	740	810	888	960	1034	1109	92
18,000,000	310	388	465	541	619	696	775	852	930	1006	1042	1162	93
20,000,000	345	430	518	602	689	775	861	946	1033	1120	1205	1290	93
22,000,000	375	458	560	655	750	843	990	1030	1123	1266	1310	1406	94
25,000,000	426	532	639	745	850	959	1065	1173	1278	1381	1491	1596	94
30,000,000	505	631	800	885	1010	1138	1262	1390	1519	1640	1772	1895	95
35,000,000	590	738	885	1033	1180	1325	1478	1622	1772	1918	2063	2219	95
40,000,000	668	833	1000	1168	1335	1500	1669	1858	2025	2165	2340	2500	96

This table is made up by taking the column of efficiencies from record and experience, and as the water end horse power is a positive matter governed by the water pressure and the quantity of water, the indicated steam power to meet the efficiency stated, is found by dividing the water end horse power by the given percentage in the table; as, for example, if the water end horse power is 400 and the efficiency 89 per cent the indicated power will be 400 divided by .89, and the result is 450 in round numbers as the steam indicated horse power of the pumping engine. The scale of per cent efficiencies in this table will no

doubt be considered low by some, but it is safe and conservative and provides against a lack of steam power in the machine, which is an enemy to good economy. If any builder of pumping engines can produce machinery with a higher mechanical efficiency than that shown in the table, and some undoubtedly can, in the lower ranges given, then so much the better and safer for long continued results.

As this chapter deals necessarily to a considerable extent with the boiler question, it will not be out of place perhaps to refer somewhat more concisely to that portion of a pumping plant. The higher and higher steam pressures which have gone hand in hand with the greater and greater steam economy of the last ten years or so have changed ideas on boilers, brought greater horse power per boiler by enlarging the pumping unit and gross demand, and has lead to restricting the dimensions of the boiler plant so far as practicable in proportion to the power developed. Probably for regular good every-day efficiency, the horizontal return tubular is as good as any, and better than most; but, where large powers are involved, high steam pressures used, the room required considered, and including the size of the necessary buildings, a limit is placed upon the consistent size of the boilers and units of this type of steam generator. The writer does not look with favor upon underfired boilers with shells of large diameter; and, although special and comparatively expensive plants, the designing of which has been in the hands of thoughtful and dexterous engineers, seem to demonstrate the feasibility of using such boilers occasionally, still in the long run and among the many plants built, the boiler for high pressure, made up of parts of comparatively small diameters upon the sectional or unit principle, seems to economize space, buildings, first cost of the completed plant, and other important particulars in that line, to a very satisfactory degree. And therefore the writer takes the ground that under present circumstances at least, considering unit capacity, gross demands, economy of construction, convenience, and economy of operation, together with considera-

tions as to buildings and space required, the water tube boiler fitted with automatic stokers, takes the lead as a steam maker for water works pumping plants.

Whatever there is in economy from utilizing waste heat in the back flue of the boilers is a credit to the boilers and not to the engine, and of course reheaters for the receiver steam can be provided and this steam made a vehicle for the transportation of the heat getting away up the chimney, if such be the case, back to the engine, and made to do useful work there. This is no special credit to any particular type of engine beyond presenting facilities for the use of heat which the boilers are allowing to escape. But it will reduce the coal bills by turning into useful work some of the heat of combustion unabsorbed by the boiler heating surface.

It is not entirely clear why more of this practice of flue reheating has not been done. It certainly has been known long enough. The writer has now before him a supplement of the *American Machinist* of October, 1878, illustrating the Pawtucket pumping engine designed and built by Geo. H. Corliss, in connection with which flue reheating was very successfully used to the surprise and confusion of the experts, who found from indicator cards that more steam apparently came out of the low pressure cylinder than there was to go into it, before it was discovered that some steam was added to the high pressure exhaust by the boiler flue turning the condensation into live steam. In this case, the duty given by a small cross compound engine reached the very satisfactory figures of a little over 133,000,000 ft. lbs. with 100 lbs. of coal. Within the past five years the Barr Pumping Engine Company used this device at Haverhill, Mass., also in connection with a moderate sized cross compound pumping engine, and the very best of authority reports over 150,000,000 ft. lbs. duty per 1,000 lbs. of steam, which at 9 lbs. evaporation, a fairly good figure for high pressure boiler work, would amount to 135,000,000 ft. lbs. per 100 lbs. of coal.

Superheated steam and generally higher steam pressure up

to about 175 gauge pressure will take place gradually, and that will likely mark the desirable and practicable limits of the present record-making plant, with the vertical reciprocating steam pumping engine. The present type will hold this line of advance stubbornly, and it will require a great deal more progress than is evident in any direction at the present time, 1907, to dislodge or even shake it materially. Its capital and fuel accounts even with coal at a moderate price make a very satisfactory showing just now, and the yearly maintenance account in the presence of good design and construction will not exceed 5% for boilers and 2% for machinery, in moderate and good sized plants where it is reasonably well cared for. So far as fuel economy alone is concerned, the present day gas engine is rapidly coming to the front, but there are several mechanical details to be carefully and successfully thought out, before gas will be used for pumping water to any great extent.

Therefore it looks to be fairly safe to plan away for another decade at least, perhaps much longer, and keep an eye upon the gradually increasing size of the pumping units during the remodeling of old plants and the construction of new plants. The perfect plant for water works pumping so far as present evidence goes, will apparently involve the following items:

Water tube boilers.
Mechanical stokers.
Natural draught, at least 0.8 of an inch of water.
Feed water economizer heaters.
Automatic damper regulators.
Coal bought upon the basis of heat units and quality.
One hundred and seventy-five lbs. steam pressure per gauge.
Moderately superheated steam by independent apparatus.
Modified steam jacketing and reheating.
Smoke flue reheating.
Vertical triple expansion pumping engines of long stroke.
Maximum piston travel, 200 ft. per minute.

Maximum rotative speed, 20 revolutions per minute, or equivalent cycle.

Coal per indicated horse power per hour, 1 lb. for large plants.

Coal per indicated horse power per hour, 1.75 lbs. for small plants.

Maintenance of engines, 1.5% for large plants.

Maintenance of engines, 3% for small plants.

Maintenance of boilers, 5% for all sizes.

These and some other items of a similar nature are about what a look ahead discovers as the coming events in the planning, construction, and operation of pumping plants for municipal water works.

CHAPTER IX

THE WORTHINGTON DUPLEX PUMPING ENGINE

As regular standard pumping engines, each kind repeatedly produced practically alike, and installed in regular water-works pumping stations, there have been only four which have been built to an extent and in sufficient numbers to fairly entitle them to the distinction of being called types. These are as follows:

The Worthington compound condensing duplex pumping engine, 1863.

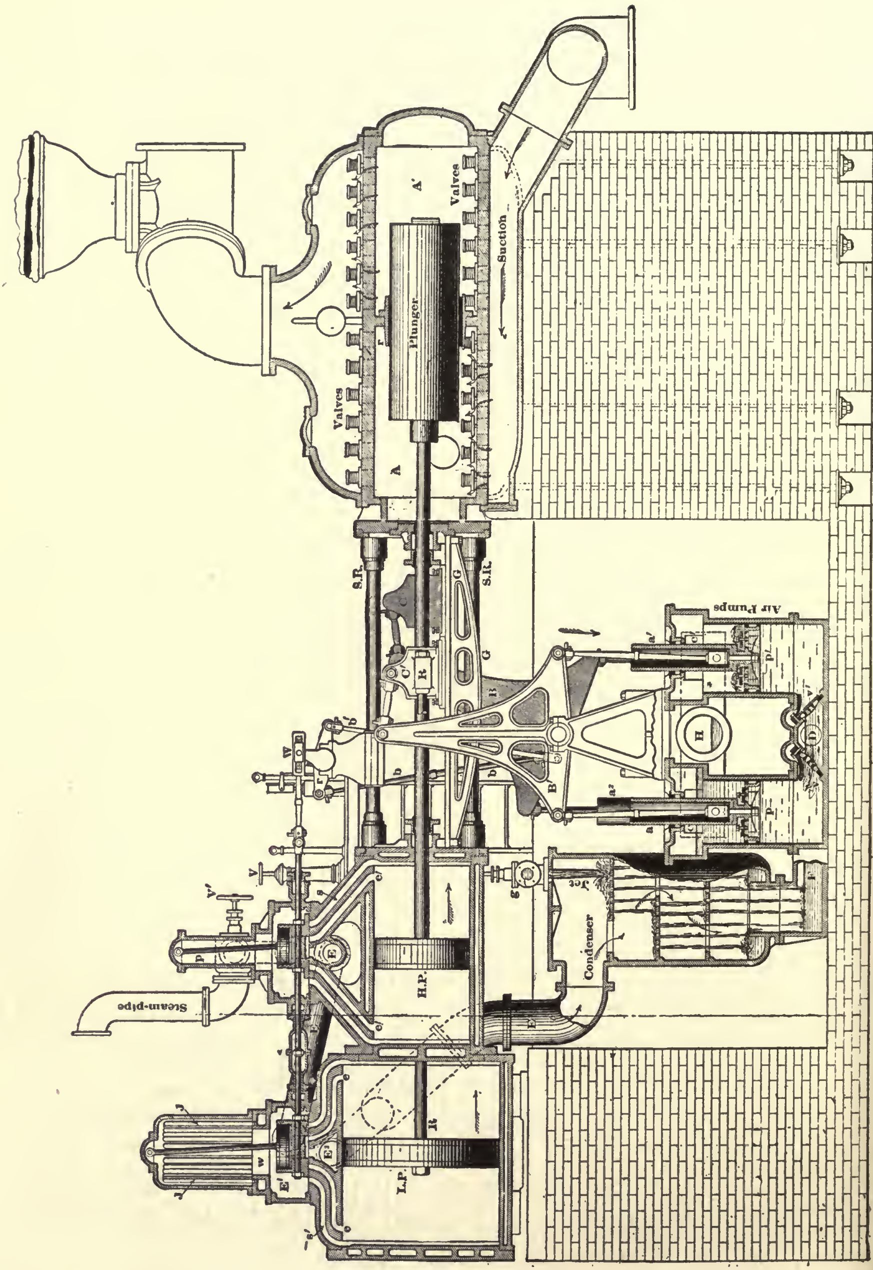
The Holly quadruplex compound condensing pumping engine, 1872.

The Gaskill compound condensing pumping engine, 1882.

The Reynolds triple-expansion condensing pumping engine, 1886.

The first of the pronounced types, the Worthington duplex, non-rotative water-works pumping engine, began to appear in 1863, designed by Henry R. Worthington of New York, and has been looked upon by engineers of high standing as one of the remarkable inventions of the times. It was at first built exclusively as a horizontal engine, but to meet later and present day requirements and demands has also entered the field of vertical machinery. Within 10 years of its first appearance, say by 1873, it had been reduced practically to a repeated and standard type; and as a compound condensing, duplex, and so-called "low duty" engine, had been installed by 1880 in about 90 pumping stations, with an aggregate capacity of over 400,000,000 U. S. gallons per day. (See Fig. 15.)

The machine consists of two complete pumping engines, simple or compound as the case might be, placed side by side

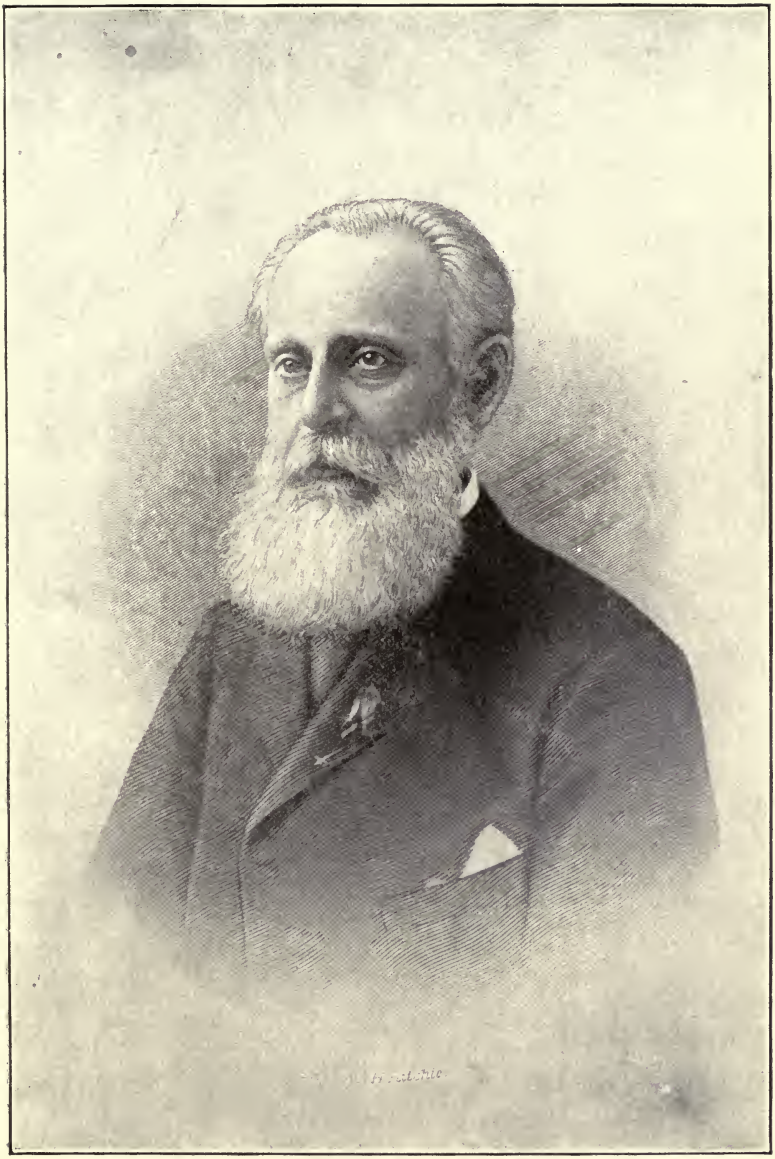

Plunger
Valves
Valves
Suction
S.R.
S.R.
Air Pumps
Condenser
Jet
Steam-pipe
H.P.
L.P.

with their center lines parallel with each other; each engine working the main steam valve of the other; these main valves being of the plain slide variety and generally fitted with means for balancing the steam pressure; but when cut-off valves are used, each engine operates its own cut-off mechanism. The steam jacket, the condenser, the air pump, and other primary features introduced by Watt were freely employed in the design of this engine. It is the simplest form of machinery with which water can be pumped on a scale corresponding to public water supply service, and has only the necessary steam pistons with which to utilize the heat energy of the steam, and the necessary water plungers with which to force the water against the pressure of the hydraulic load, together with light and simple connecting bars for keeping the proper relations between the steam and water ends of the machine. It has no massive framing, no heavy shafts and fly-wheels; only some insignificant looking connections for operating the steam valves which distributed the steam to and from the places where it did its work. It was handicapped in its low duty form for this day and age by the fact that it could not expand steam beyond certain limited ratios, and if it could have retained its simplicity of a quarter of a century ago, and at the same time greatly increased its steam economy, there would have been to-day but a limited place for the crank and fly-wheel type of pumping engine.

Each pair of pistons, one high pressure and one low pressure of each of the tandem compound steam ends, is coupled to a double acting water plunger by an extension of the high pressure piston rod to a cross head, into which is keyed the plunger rod leading into the water cylinder and so driving the plunger directly from the steam pistons without the intervention of any kind of mechanism or machinery; hence the name "direct acting" which was first, and is yet, often applied to this type of pumping machinery. The front head of the low pressure cylinder and the back head of the high pressure cylinder, are formed practically in one piece, the high pressure being in front

of the low and close to the latter; which is to say that the high pressure cylinder is situated between the low pressure cylinder and the water end, the low pressure cylinder and the water end resting directly upon the foundations, and the high pressure cylinder partially supported by an iron column and partially from the low pressure cylinder head. The low pressure piston has two rods, which pass through long sleeves or really extensions of the low pressure stuffing boxes, outside of the high pressure cylinder barrel. The low pressure piston rods extend and are keyed to the same cross head which receives the high pressure piston rod, thus making an arrangement of three steam piston rods at one side of the cross head, and a single plunger rod at the other side; a very compact, strong, and satisfactory form of construction. In the very early and quite large engines, the high pressure and the low pressure rods were coincident with each other, forming a single steam rod passing from the low pressure through a closely fitting sleeve in the front head of the low pressure cylinder and so into the high pressure, then out of the front head of the high pressure, and so on to the main cross head, to which, at the other side, the plunger rod was secured.

The steam and water ends of the engine are connected by very simple framing or cradle bars, as there are no pillow blocks or crank shafts to provide for. In the early engines these frame or cradle bars were simply four heavy turned and polished, wrought-iron rods, extending from the high pressure cylinder heads to the water ends or cylinders, and upon the upper pair of bars there attached the bearings for the valve gear. In later engines the framing is composed of heavy cast-iron cradles to which are attached the steam valve mechanism and other necessary details. The finished wrought iron cradle bars connecting the steam and water ends, the steam cylinders handsomely lagged with dark-colored wood, and the clean cut, well-designed water cylinders, formed a substantial, neat, and altogether handsome and attractive piece of machinery. No doubt the early Worthington pumping



HENRY R. WORTHINGTON.

engines of goodly size, constructed as above, formed a type of machinery, which for grace and appropriateness, has never been surpassed in the line of water-works pumping machinery. The writer was at the Newton, Mass., pumping station in July of 1905, where one of the old time Worthington pumping engines of 7,000,000 U. S. gallons capacity was at work, and has been at work over 27 years pumping water for public supply, and was at the time of this visit working against a water load of about 100 lbs. pressure. No quieter, smoother running, or more satisfactory machinery could possibly be found for the purpose of pumping water, and it is now only outdated on account of the fuel burned for regular work, but is held in reserve when other machinery requires attention. This early type of pumping engine has always been noted for its smoothness of action in the work of pumping against a heavy head and under high pressure loads, and seems to work as perfectly and with as little noise as under lighter heads; this no doubt being largely due to the fact that the engine is at all times almost completely under the control of the water column, the engine seemingly doing nothing further than to furnish the power necessary to overcome the water pressure, while so far as the flow of the water is concerned, and the actual timing of the movements of the plungers and pistons, the hydraulic action is in control of the main pumps.

The main pumps are of the inside ring type, secured directly in line with the steam cylinders, so that the force and resistance of the pumping is in the same center line, making the engine self-contained so far as concerns the work to be done and the material used in construction. The outlet from the force chambers are 90° bends turned towards each other and are joined by a three-way casting upon which is mounted the air chamber, only one air chamber being used, but it is situated at a very advantageous point with reference to the delivery of the water alternately from the two water cylinders. From the outer opening of this three-way casting, the force main begins, and is lead down through the floor and so out of the

building. The pump chamber, or water cylinder proper, has at the middle of its length an inside flange to which is bolted the ring or sleeve through which the plunger works, and so does the pumping. The valve decks or flat surfaces, properly ribbed for strength, are situated in horizontal planes directly above and below the water plunger, and the brass valve seats are either screwed into, or are bolted onto the valve decks; the valves, see Fig. 16, are rubber discs of moderate or small size, moving vertically upon brass stems and are controlled either by weights, as in the early practice, or by springs as in the later work. Immediately below and above the water cylinder are situated the suction and force chambers, respectively; the former joined by a cross suction with easy bends for the flow of the incoming water, and at the outer extremity of the cross pipe is located the main suction pipe which usually drops vertically into the pump well from which the engine draws its supply. The suction pipe is sometimes extended to the outside of the building, and the machine pumps from a well located at some convenient point near the walls, or not to exceed, say 30 feet preferably, although the writer has located and used pump wells 60 feet away from the building with satisfactory results, provided the size of the

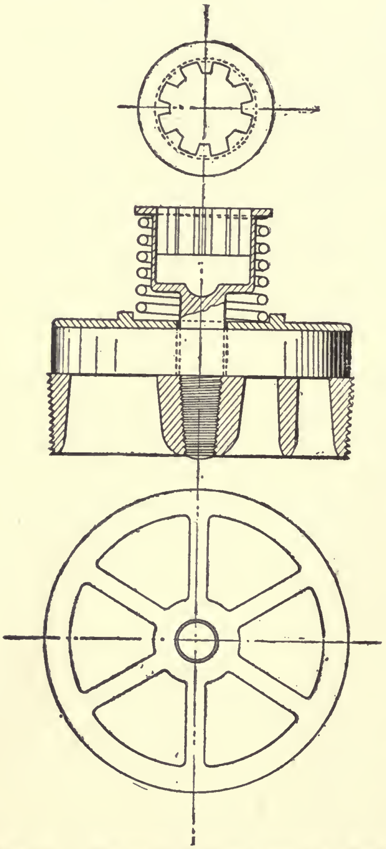


Fig. 16. — Standard Rubber Valve and Seat.

suction pipe was large enough to keep the friction of flow down to a point amounting practically to nothing, or at least an acceptable minimum.

The force chambers are bolted to the upper surface of the water cylinders which form the discharge valve decks, and from the middle of the length of each force chamber the water is discharged through an appropriate opening formed into a flange for the reception of the 90° bends already referred to. The force chamber, the water cylinder, and the suction chamber are provided with handholes and manholes for inspecting and giving proper attention to the interior of the pump when occasion requires, these holes being closed during operation by bolted plates, the largest ones being provided with hinges to facilitate handling. The flow of the water into and through the suction chamber and valves, directly upward through the discharge valves and force chamber, and so on into the force main, with the liberal dimensions employed in this type of engine even 40 years ago, leaves nothing to be desired in the way of low rate of water friction and high percentage of hydraulic efficiency.

The tandem compound, steam end of the engine, is a regular reciprocating steam engine, with steam jacketed cylinders, with pistons working in couples, one high and one low pressure, and with plain balanced slide valves located at the top of the cylinder barrel. The pistons work in the same axial line and the valve seats are in the same plane, the high pressure seat being raised the difference in the half diameter in the two cylinders, so as to use a practically continuous valve stem for both main valves; the balancing piston and supporting column is directly over the steam chest; and the entire design has proved itself by years of usefulness to be of the very highest type of means to suit the ends sought.

The air pumps, four in number generally, although sometimes two are used, as, for example, in the Newton engine, are placed in a small pit formed by the space between the water and steam ends of the machine, and are driven by a tri-

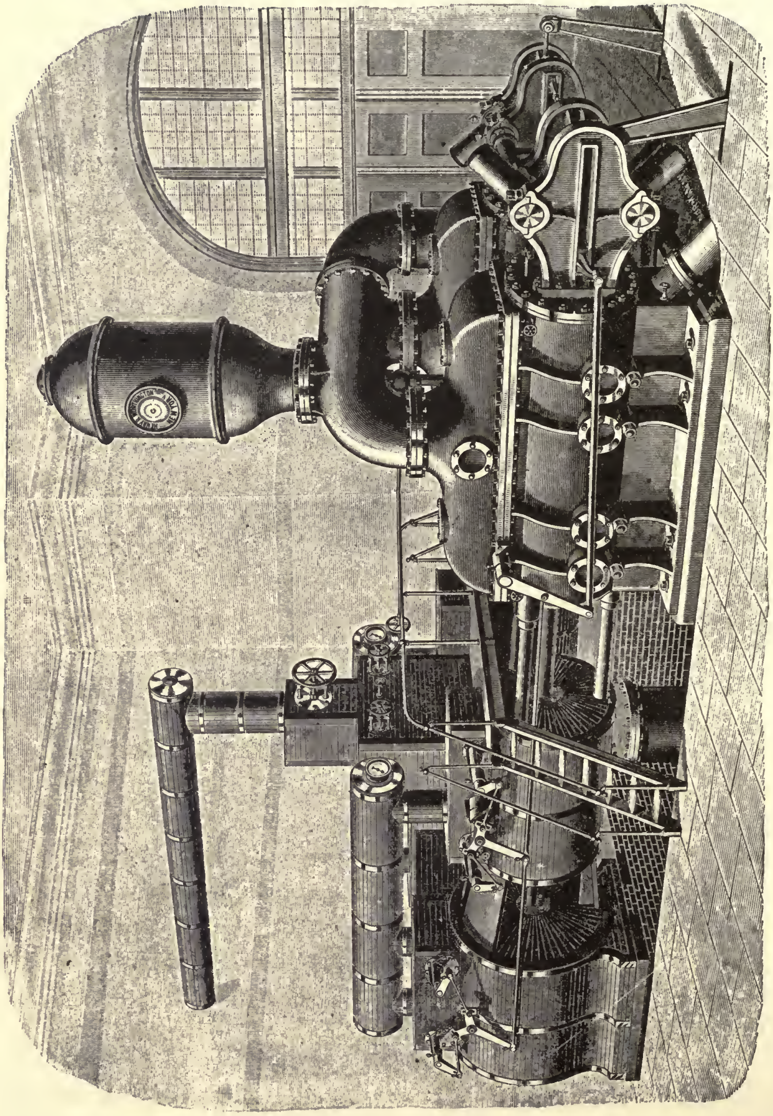


Fig. 17. — Worthington High Duty Horizontal Compound, Perspective.

angular bell-crank lever connected to the main cross head, and the shafts of the bell cranks are utilized for giving motion to the valve gear. The condenser is situated directly beneath the high pressure cylinders and half way between the two, upon the lower portion of the foundation, making it possible to have a very short exhaust pipe from the low pressure cylinder.

The steam pistons are simple and of the well known cast-iron ring packing variety, with generous bearing surfaces, as is demonstrated by their long life and usefulness in every-day operation. The water plungers are generally hollow and aim to be water-tight, so as to secure partial flotation, but it is doubtful if they remain empty any great length of time, and indeed it matters little whether they do or not, as the friction and wear could not be a very material matter in any event. This form of plunger will keep practically water-tight through the sleeve or ring, or as water-tight as most plungers in average clear water for long periods of time, as is evident from known results. One case known to the writer was at Toledo, Ohio, where the engines had been on stand-pipe service for 12 years, which is really direct service in the sense that there is no storage, and the machinery must run continuously; the water was not clear but contained a fine clay silt extremely difficult of sedimentation, and upon test the plungers leaked so little under 80 lbs. pressure with one pump head removed, that it was difficult to make accurate measurements of the quantity.

The original Worthington pumping engine possessed the qualifications for the first duty of a pumping engine to "PUMP WATER," and to do so continuously with the least practicable interruption, to a degree which has never been equaled by any machine ever produced for the purpose; and not even by the improved or high duty engine of the same type. It holds the record for durability. But its steam economy was limited to the possibilities of expanding steam at full stroke through two cylinders of different diameters alone, and without the high expansion ratios possible with the use of cut-off valves. It had apparently reached the economic limit with compound

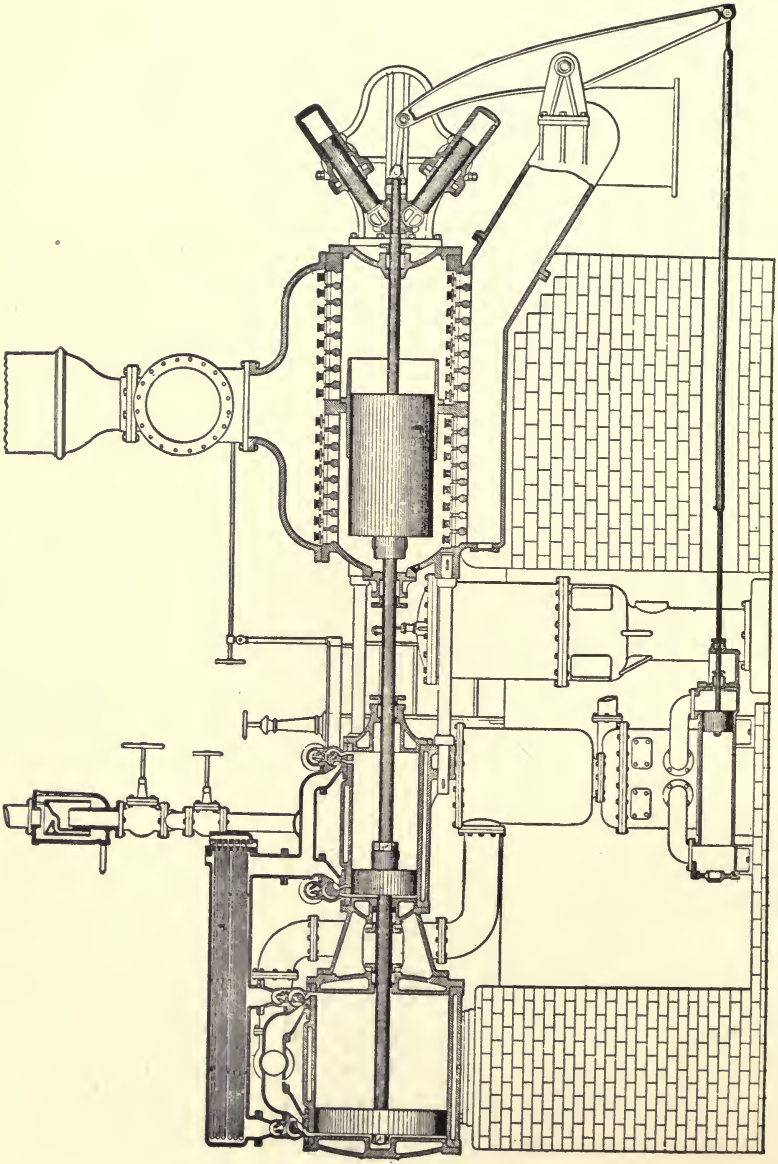


Fig. 18.—Worthington High Duty Horizontal Compound, Sectional.

cylinders alone without cut-off, and with the steam pressures usually employed. But, of course, the question arose naturally enough in time, can this machine, so satisfactory on the

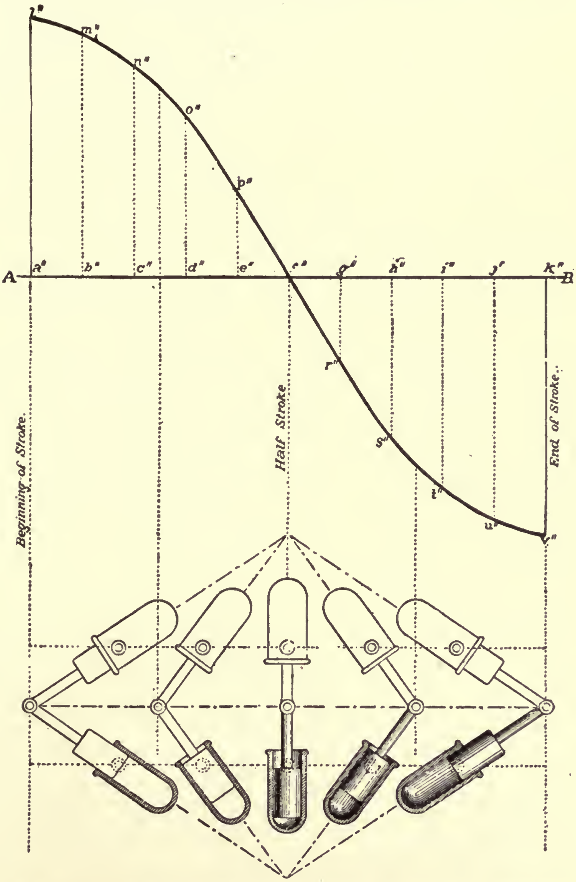


Fig. 19.—Worthington Compensating Cylinder Diagram.

main point, that of certainly pumping water, be improved in its economy of steam without giving up its best features of design? And this question has been answered by the introduction of the high duty Worthington engine, in 1886, as shown in Fig. 17 in perspective, and in section in Fig. 18, although

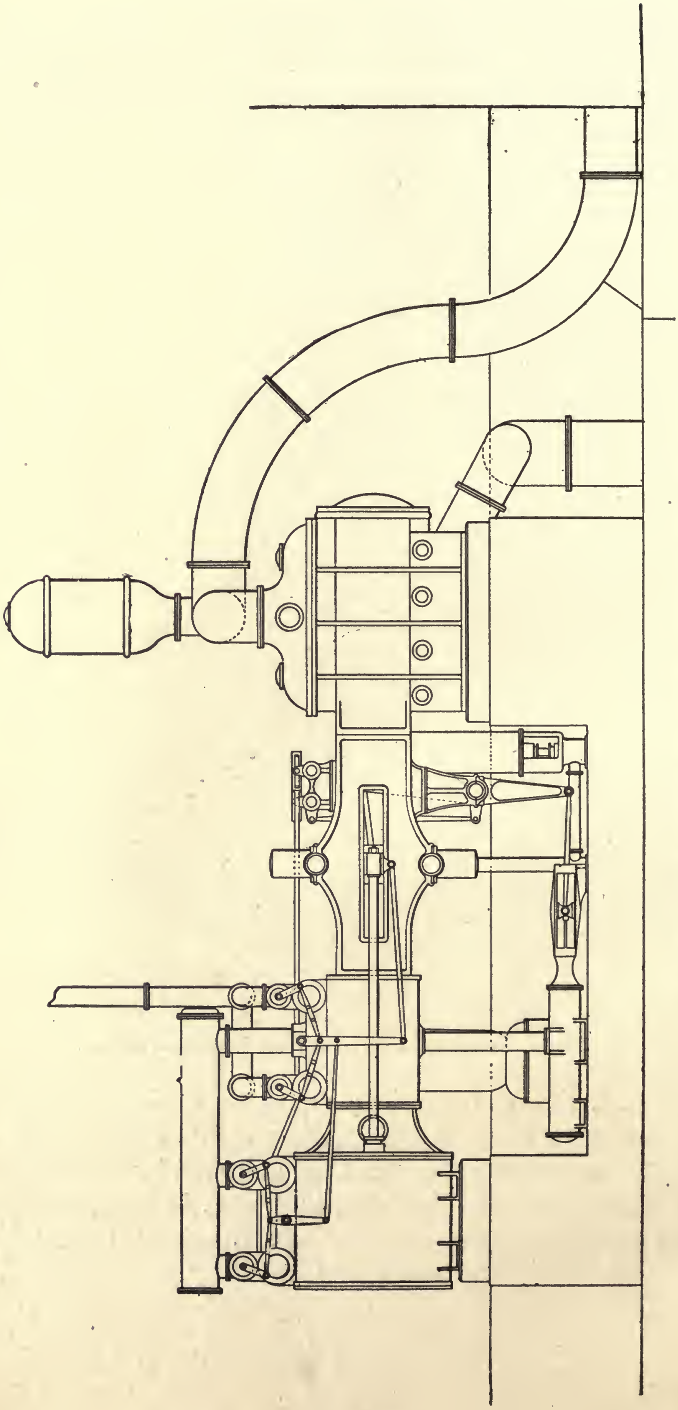


Fig. 20. — Worthington Horizontal Compound, High Duty.

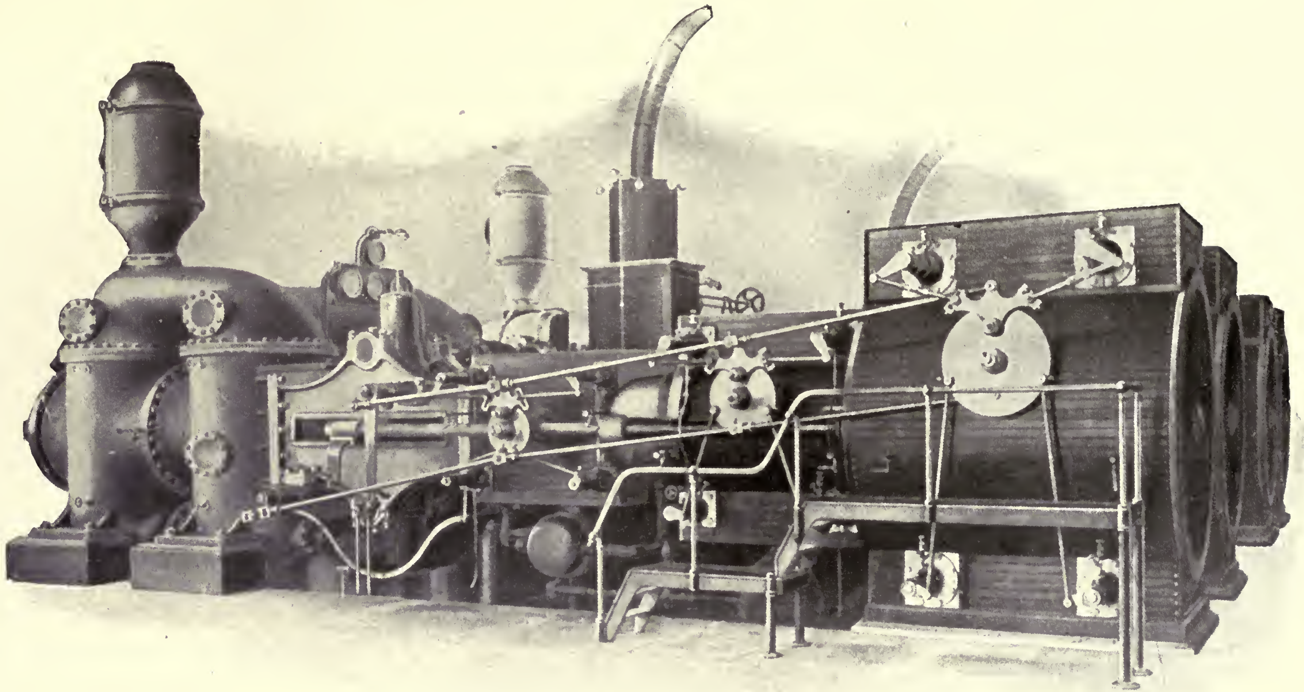


Fig. 21 — Worthington Horizontal Triple, High Duty

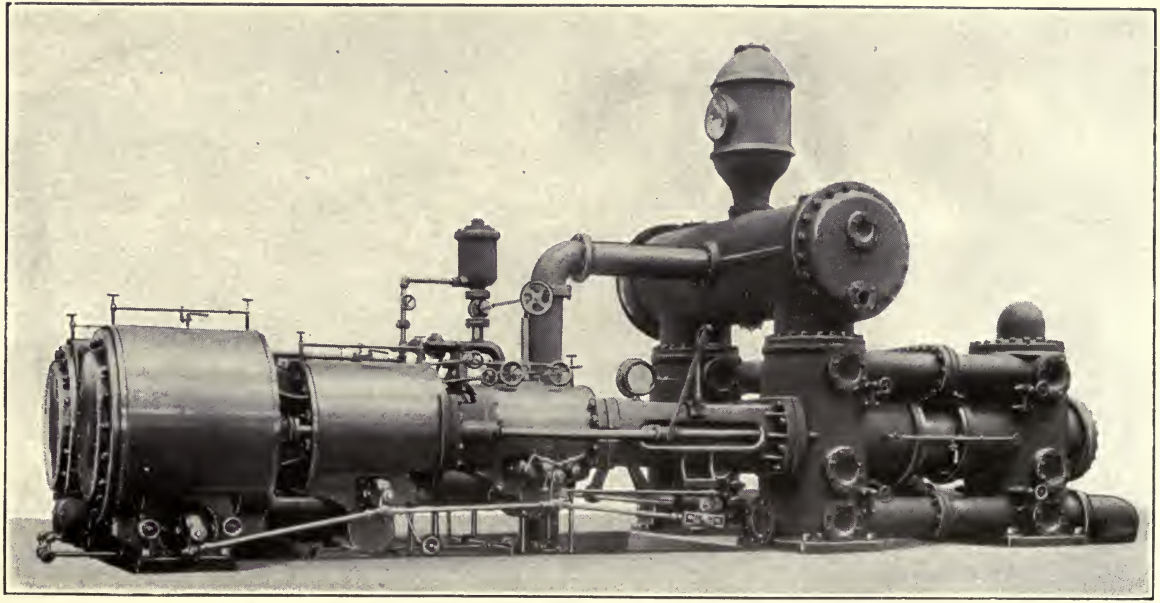


Fig 22. — Low Duty Worthington Triple, Horizontal

some of the simple effectiveness has been sacrificed, but not seriously so. And so it turns out that the argument of low first cost and low duty, against the argument of higher first cost and higher duty, has been nullified by simplifying and reducing the cost of the crank and fly-wheel pumping engine; and by complicating and increasing the cost of the non-rotative or direct-acting pumping engine; both results being forced upon each type by the struggle with each other for commercial supremacy, and the final result being that both types now meet on common ground as to the commercial situation.

In the newer Worthington engine the most prominent features remain the same as in the older engine, as to general arrangement of steam and water cylinders, but a device has been added which is called the high duty attachment and consists of a pair of small, oscillating water cylinders attached to some place in the framing, sometimes beyond the outer ends of the main water cylinders, and sometimes between the steam and water ends of the engine. Connections are made through the trunions of these oscillating cylinders with a hydraulic accumulator, which in turn is connected with the water pressure in the force main, and by this means the pressure on the inner ends of the plungers in the oscillating cylinders is maintained in close relation with the pressure in the forc main against which the engine is working. These plungers with their swinging motion, which constantly changes the angle with the center line of the engine, resist the advance of the steam pistons at the beginning of the stroke just when the initial steam pressure is too much to drive the water plungers against the load, and helps the steam pistons finish their stroke when the expansion has brought the steam pressure down to a point which is too low to drive the water plungers without some kind of assistance. In short, the oscillating plungers change from resistance to assistance during each stroke of the engine to meet the varying pressure given out by highly expanding steam, and which the older engine did not have to struggle with. Fig. 19 shows a diagram which represents

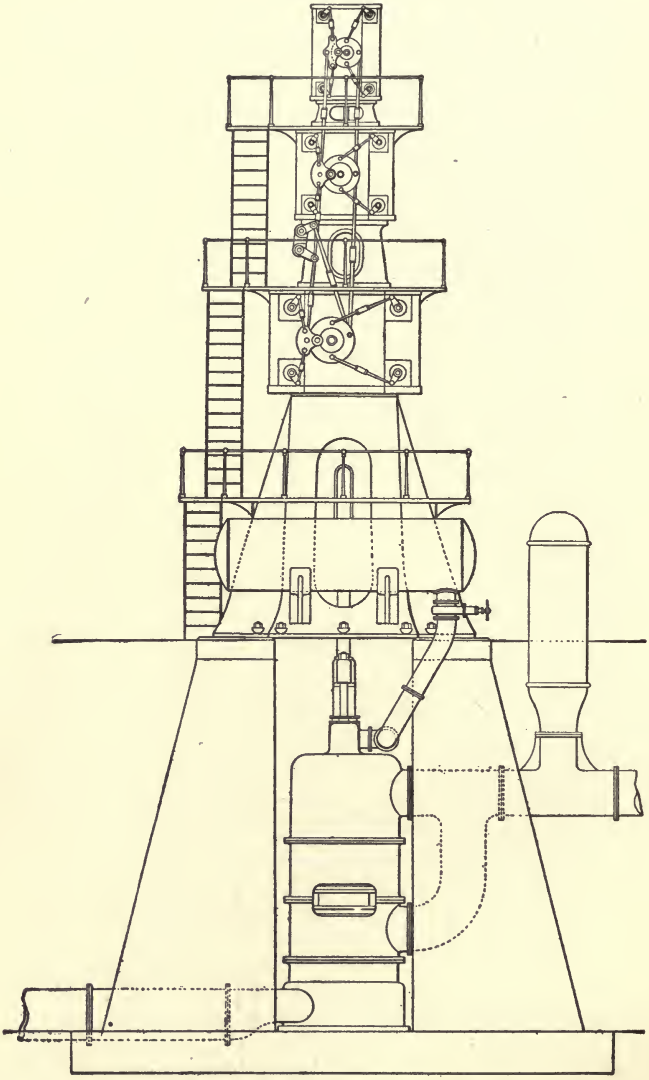


Fig. 23. — Worthington Vertical Triple, High Duty.

this action of the oscillating or compensating plungers and cylinders. Low expansion, as in the older engine, meant low duty, that is, in the neighborhood of 65,000,000 ft. lbs. per 1,000 lbs. of steam; and high expansion, as in the newer engine, means high duty, that is, 100,000,000 ft. lbs. per 1,000 lbs. of steam, and upward. The resistance and assistance given out by the compensating cylinders is just what the fly-wheel gives to the crank engine, only in one type it is done with water pressure, while in the other type with the weight of cast iron. Each pair of oscillating or compensating cylinders are directly opposite the engine center line and therefore act in concert with each other, and relieve the cross head in which they work in sockets of any cross frictional resistance or unbalanced lateral strains.

The alterations from the old to the new were cautiously made, and at first with as few changes as possible; and even at that, unknown difficulties in construction were bound to be met in such an unexplored field. At first the plain slide valve on the steam cylinders was retained, and cut-off valves added. Then the large slide valves were discontinued and what might be called the Corliss type substituted, the first important high duty engine being a 15,000,000 gallon machine for Chicago in 1889, with all of the valves placed at the top line of the horizontal cylinders, using four valves, but having two main valves for induction and exhaust, and directly over these, two cut-off valves, which cut off the steam after the main valves had admitted it, and then drew back out of the way in time for the next admission by the main valve. The exhaust was accomplished through a cavity in the under side of the main valve, the same practically as in a plain slide valve.

This change in the steam valves immediately reduced the clearance or waste room, removed the necessity of having the valves in the same plane because radius rods were submitted for ordinary valve stems as before, and improved the steam economy. Finally the four valves were placed at the four corners of the cylinder, as in the drop cut-off arrangement,

and were worked by a sort of compound wrist plate for which a motion was obtained peculiar to the direct acting machine. The operation of these steam valves was obtained by positive motions, as it was clear that a releasing and drop cut-off would not be safe or regular on this type of engine. Any point of cut-off can be secured by changing the location of a sliding block; the operation is without noise, and the various levers, rods, and pins have easy work and consequently little wear.

Fig. 20 shows a compound condensing, high duty, horizontal Worthington engine, with attached air pumps and jet condenser, accumulator, compensating cylinders, valve motion, reheaters, and other details. A horizontal triple expansion engine is shown in Fig. 21.

A vertical triple expansion engine of 20,000,000 U. S. gallons daily capacity is shown in Fig. 23, and exemplifies the very great departure which time and the demands for larger units, high duty, and other considerations have brought about since the early engines of the low duty horizontal type were built and held the front of the stage for water-works pumping machinery.

The Worthington water-works pumping engine has also appeared as a low duty triple expansion machine, which consists of the original compound engine with an additional pair of steam cylinders attached, but without the compensating cylinders; the added cylinders being generally in front of the former high pressure cylinders, thus making the new ones the high pressure, the former high pressure into intermediate cylinders, and with the low pressure cylinders remaining as formerly. This form hardly approaches the dignity of a distinct type but rather should be called a class, because it involves the principal features of the real type. It is only applicable to moderate capacities, it generally paying better to have a regular high duty compound when the required capacity gets up to say 5,000,000 gallons per day and over. A cut-off is usually applied to the high pressure cylinders and then a rather high rate of reciprocation employed so as to get the benefit of the momentum of the moving parts to finish the stroke, which of

course suggests that the direct acting engine depending purely upon the force of the steam is slightly invading the field of the crank and fly-wheel machine, and that perhaps a simple cross compound engine of the wheel type might be just as well for the work. This low duty triple expansion Worthington pumping engine is shown in Fig. 22, which indicates the general features and many of the details of this form of the well known direct acting machine.

Up to 1886, when the introduction of the high duty engine had begun, the original Worthington type of water-works engines known as low duty engines, had reached 250 in number with an aggregate daily capacity of about 1,000,000,000 U. S. gallons per 24 hours, the engines varying in capacity from 500,000 to 25,000,000 gallons per day with an average of 4,000,000 gallons per 24 hours.

From 1886 to 1906 there have been produced about 225 pumping engines of the high duty class, with an aggregate capacity of 2,250,000,000 U. S. gallons per 24 hours; or about 225 engines with an average daily capacity of 10,000,000 gallons, and varying from 3,000,000 to 40,000,000 gallons per 24 hours.

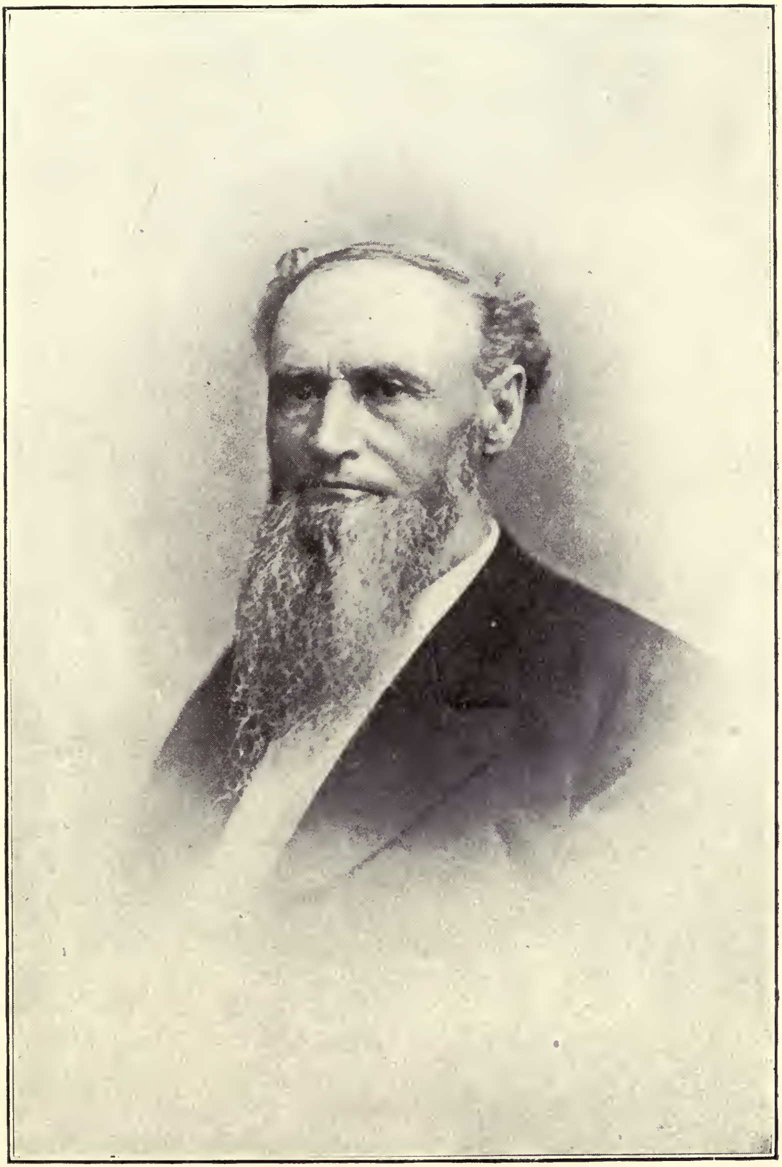
CHAPTER X

THE HOLLY QUADRUPLEX PUMPING ENGINE

The Holly quadruplex pumping engine was designed to meet an existing but a novel want at the time of its first construction. A direct system of water supply had been built and put into operation, in which water was distributed through pipes laid in the streets for the purpose of protection against fires; and by a direct system is meant a closed circuit of water pipes wherein the water pressure is raised and maintained at any certain pressure, and where there is no storage of water beyond what is contained in the pump well and the supply flowing to such well.

About 1866 the first of these direct systems was established at Lockport, N. Y., by Birdsill Holly, with a mile of cast-iron water pipes, and a rotary pump operated by a water-wheel, the plant designed for fire service only. Two years later a similar system was built at Auburn, N. Y., in which a domestic supply was combined with the protection against fire; both these systems being driven by means of water power. Of course the enlargement of such an important idea soon carried it beyond the reach of a single means of applying power, and that means such a restricted one, so that before very long the question of the application of steam power which could be located almost anywhere, came naturally enough into the proposition. And steam having once entered the waiting door, the study of fuel economy was of course inevitable.

The first attempts at steam power for this kind of water works, somewhere late in the sixties, were in the line of a steam engine driving a shaft, and from which there were driven a series of gang pumps consisting of six single acting pumps



BIRDSILL HOLLY.

arranged to operate in regular succession, the idea of units or increments applied so that the lack of uniform delivery perfectly natural to the domestic supply could be practically met, seemingly an early detail of this system and properly so, as later experience fully demonstrated. In conjunction with these gang pumps there was used a rotary pump known as the Holly Rotary Pumping Engine and Fire Pump; this fire pump being used only as an auxiliary in case of fire beyond the capabilities of the regular gang pumps. Over twenty water-works were fitted out with the gang and rotary pumping machinery, and at that early day in the development of public water-works no doubt provided good and ample service, considering all of the circumstances and the unknown road reaching out ahead of the investigator, inventor, and investor. The first duty of the pumping engine was to pump water, of course; for with plenty of water, danger could be avoided even though accompanied with moderate inconveniences. But the engine had to do some other things which were not completely appreciated by competitors who looked at the single item of pumping. And so it came about that economy of steam and fuel went hand in hand with accommodation to the conditions, and so improvement after improvement was introduced, and the next step after the gang-rotary combination was the introduction of the Holly Quadruplex Pumping Engine for the closed system of water-works distribution, in 1871 and 1872, which was especially adapted to both domestic and fire service to a very remarkable degree, and superseded both the gang and rotary pumps. The first of this type was placed in the Dunkirk, N. Y., pumping station, and was designed as an engine with four straight condensing steam cylinders; but in 1874 the machine had taken a step forward and appeared at Rochester, N. Y., as a regular compound condensing pumping engine of 2,000,000 gallons daily capacity.

The Holly quadruplex pumping engine is a crank and fly-wheel engine, probably so named to indicate a machine a point or two better than the duplex machine then in the market

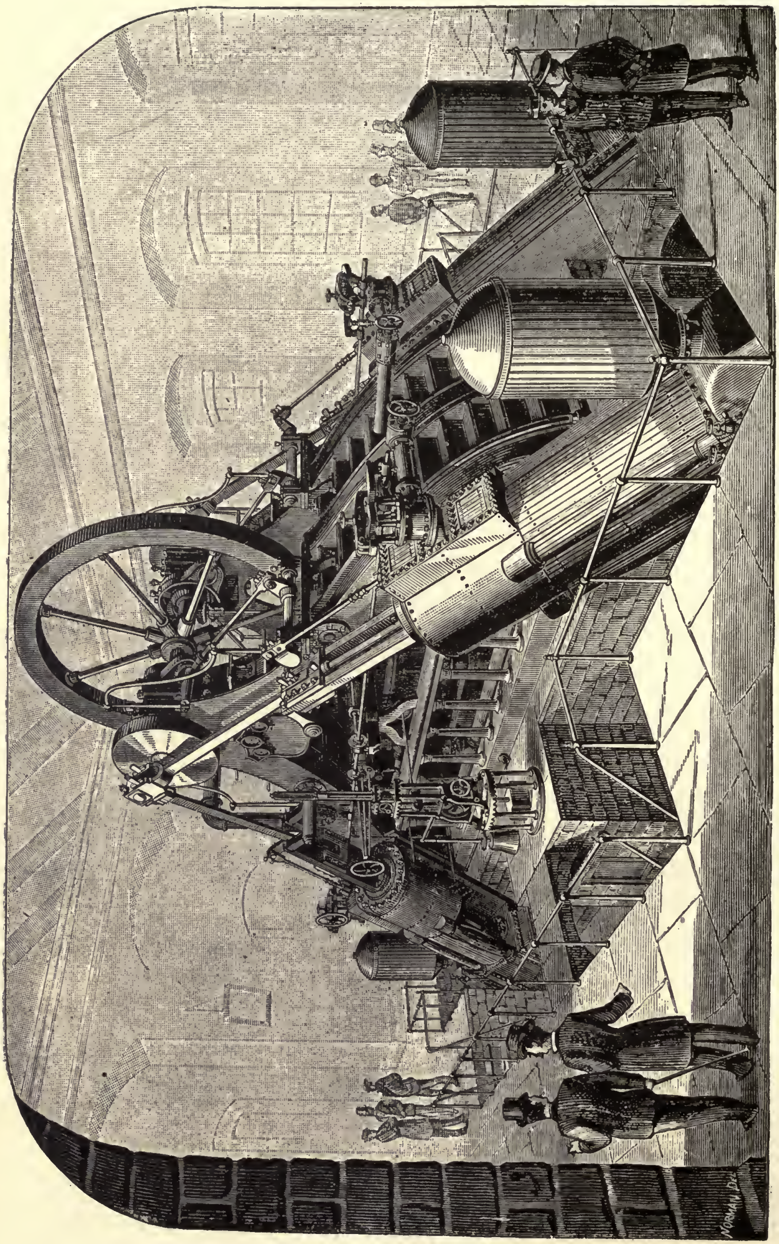


Fig. 24. — Holly Quadruplex Pumping Engine, Perspective.

at least in the mind of its inventor; it is shown in perspective in Fig. 24, and in outline sectional elevation in Fig. 25. There are four steam cylinders, the center lines through which are at an angle of 45 degrees from the horizontal; and four pumps, one of the pumps below and in line with each steam cylinder. The general or main framing is heavy and strong, and is formed somewhat like a letter A, one frame at each side of the engine. At the top or point of this A-frame is located

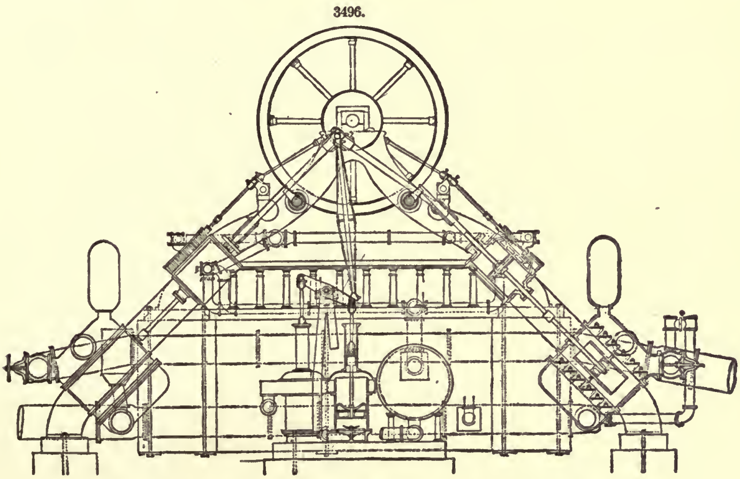


Fig. 25. — Holly Quadruplex Pumping Engine, Sectional.

the main pillow blocks or bearings for the crank shaft, and this shaft extends across the machine from one frame to the other, carrying at its mid-length the fly-wheel which is made of moderate weight so as to respond quickly to demands for changes in the speed of the engine and the variations in the quantity of water to be pumped, as this type of pumping engine was especially designed for the "direct system," sometimes called the "Holly system," wherein, as already pointed out, there exists no storage of water, the entire supply with all of its fluctuations being delivered into the distributing system according to the demands of the moment.

This engine was mostly used in the flat portions of the middle western states where no hills existed for storage and distributing reservoirs, or, where hills did exist, the cost of a reservoir was deemed out of the question, in the construction of new water-works at a time in the development of the country when such conveniences as water supplies were something comparatively new. The following description will give a general idea of this machine:

At each end of the main shaft there is secured a disc crank, with one crank pin in each crank common to a pair of the cylinders and pumps, the crank pin for one of the pair of cylinders being located 135 degrees in advance of the crank pin for the opposite pair of cylinders, so that 8 impulses are obtained at each revolution of the fly-wheel, thereby producing a very steady flow of water. This feature is especially desirable in a closed system of distribution pipes, and at very slow speed; not only for the sake of uniform pressure, but also on account of a steady rate of revolution of the engine at whatever speed was necessary, which is conducive at slow and varying speeds to the best economical results under naturally bad steam conditions. This principle of 8 impulses per revolution has been used later in large generating engines for electric railroad power, as, for example, in the power station of the Manhattan Elevated Railway, and for the New York Subway power station, where engines of 7,500 nominal horsepower have been installed; the results of the 90 degree angle of steam cylinders and the 135 degree angle of crank pins, being the reduction of the fly-wheel element, and an extremely uniform rate of revolution with very smooth-running engines. This principle of engine construction is also followed in the engines of the steamboat "Priscilla" of the Fall River Line, and the three-throw crank motion is found in the new Hudson River Day Line steamboat "Hendrick Hudson," in all of which cases steady and uniform revolutions at whatever speed the engine may be making, is desirable.

The main pumps of the quadruplex engine are of the piston pattern, secured directly in line with the steam cylinders so

that the force and resistance of the pumping are in the same center lines, thus making the pumping engine self-contained with reference to the work it has to do. Each pump has its own air chamber, situated at the highest point of the angle formed by the sloping position of the pump castings, and are most effective in their operation. The suction and force chambers are above and below the pump barrels, and the seats for the water valves are set at an angle of 45 degrees to the center line of the pump, so that when the pump is in position on the engine, the valve seats become horizontal. These water-valve seats are arranged in the deck form, one deck in each pump for suction valves, and one deck for discharge valves; the flow of the water being accomplished with very little deflection from the natural course from suction to discharge chambers.

The pump barrel proper is an independent piece securely bolted to a rib which divides the pump chamber into two portions corresponding to the two ends of the double-acting pump. The pump barrels and pistons may be readily removed, repaired, or replaced, practically without throwing the engine out of service, and as there are four main pumps it will be seen that this machine is especially adapted to direct service, and where it is impossible to have reserve engines on account of cost; it being remembered that the Quadruplex was introduced in the early stages of water-works, and very often the total cost of the system had to be closely reckoned, in order that any system could be had at all. It is pretty safe to say that no other pumping engine has been so carefully studied to adapt it to the necessities and wants of the water consumer, and to meet conditions as they actually existed in the direct service or closed system of pipes. The cross-connecting pipes for suction and for delivery were carefully arranged both as to position and necessary shut-off valves, so that any of the four main pumps could be taken out of the work or off of the engine with little or no interruption of the service. Therefore, it will be seen that the builders of the quadruplex pumping engine, instead of insisting what the users of water ought to do, or not

to, they recognized the helplessness of their position, and made the engine do what was necessary to practically meet, to a reasonable extent, all and every demand where pressure and quantity of water were involved; and in this, as time has passed, the cool and careful historian must say that they succeeded.

The engine proper, or the steam-power end of the machine, is a regular reciprocating, crank and fly-steam engine, with guides and connecting rods not unlike those of the locomotive. An auxiliary connecting rod attached to one of the crank pins operates the air pump for the condensing apparatus; the air pump being driven by means of a rocking beam and shaft, the beam not only driving the air pump but also two boiler feed pumps, one of which takes its water supply from the overflow from the air pump, and the other taking the water of condensation from the steam jackets, and sending it to the boilers.

The connections between the steam and water pistons are by means of rods and keys in the usual manner of securing such rods, but there is an intermediate coupling between the steam and water rods for facilitating the taking off and putting on to the work, more or less, of the pumps as the case might require. The steam piston has special arrangements for saving time in adjustment; the piston packing is usually of the regular cast-iron ring variety, set out by means of steel springs; and the set screws for adjusting these springs project beyond the face of the piston so that the packing rings may be adjusted through handholes in the lower cylinder head, by simply removing the bonnets. This is only one of the many devices provided for easily, promptly, and quickly meeting the inevitable manipulation necessary in all kinds of machinery, but which in the case of a public water supply must be done as nearly as possible without interruption. Now cities and towns are older and more mature, and the water works question is an old one and familiar to all; and in the natural growth, older engines forming reserves, the necessities are removed which existed forty years ago, when only one engine could be afforded, and that one was all and everything to the plant.

The arrangement of the steam and exhaust pipes is such that the steam can be utilized in several ways: Steam directly from the boilers can be admitted to all of the cylinders, and exhaust into the condenser; or, steam from the boilers can be admitted to only one cylinder, and thence exhaust into the other three, and then sent to the condenser. So that four straight condensing engines, a compound with one high pressure cylinder and three low pressure cylinders, a triple expansion, or a quadruple expansion engine, can be formed and operated at will. But on direct service, the ultra-refinements possible to reservoir work where the engine can run at full speed or nearly so, are not profitable, and cannot be utilized by taking advantage of the multiple features possible in the quadruplex engine. The pumping and distribution of water takes first place in the direct system, and extreme steam economy is secondary. Even to-day, on electric railway service, the cross compound engine is not of so high economy as the triple machine, but it is better adapted to take care of the "peak" of the load, and the better of the two in all-round performance; it is an unconscious reflection of the qualifications of the early Holly pumping engines to meet variable demands.

In the quadruplex pumping engine the steam valve gear of each cylinder consists of one slide valve, which with its steam chest and valve stem also resembles the locomotive and the early steam engines before the advent of the automatic cut-off. A double poppet valve in the steam chest is used as a cut-off valve back of and above the main valve, and is operated by means of a spiral cam, this cam being controlled in a motion lengthwise of its revolving shaft so as to vary the point of suppression of the steam entering the cylinder. The variation of admission and cut-off of the entering steam is from nothing to full stroke, and this full stroke range of cut-off is necessary to the proper regulation of such slow and variable moving pumping machinery working upon a closed circuit of pipes. The means and method of adjusting the cam to the desired point corresponding to the work needed to be done by the engine,

are very important features of the Holly pumping apparatus; and the automatic regulator provided for the purpose, in fairly good hands, or in as good hands as ought to be found in a public pumping station, seems to be excellently adapted to the work. The regulator is connected to the water main so that any change in water pressure is promptly met by a corresponding adjustment of the steam supply, resulting in a practically uniform water pressure with a widely varying supply.

This matter of regulating the speed of pumping engines, the quantity of water they pump, and the pressure in the force main, has been much studied from time to time by engineers and pumping engine builders. The writer distinctly remembers that a good many years ago he put in a lot of planning over differential and compound levers acting against a water plunger and controlled by weights, so as to produce the effect of a spring but with the use of a weight, and so produce a controllable effect as with a weight instead of an unknown increasing effect as with the unknown and unknowable characteristics of a steel spring. But the apparatus was completed and attached to the original beam pumping engines put into the then new water-works at Terre Haute, Ind. Just how it worked in actual service has passed from memory; and perhaps it better stay there, wherever it is.

The regulating of a pumping engine at work has several features quite different from the regulation of steam engines generally, very much to the surprise of steam-engine builders who thought that the only way to make a pumping engine was to construct some pumps and then attach them to their steam engine, and that would settle everything. With a mill or factory engine, for example, the aim is to keep the speed uniform while the amount of power changes; but with the pumping engine, the speed and power both change, and the aim is to keep the water pressure constant. Further, with a pumping engine with a fixed cut-off, the simple fact of increasing the speed by attempting to lower the water pressure by drawing more water, increases the power of the machine by giving it a

higher piston speed with the same mean effective pressure in the steam cylinders, and the only increase of power needed, aside from what the engine will naturally give by automatically increasing its speed, is for the friction of the increased quantity of water flowing. For example, a few years ago 85 lbs. steam pressure, 75 lbs. water pressure, and 5,000,000 gallons per day would be pretty good work to be done by, say, an 8,000,000 gallon engine working upon direct service with the necessary capacity for reserve; and to make the illustration simple, the work is supposed to be done in one straight condensing cylinder, although the effect would be the same distributed among the four cylinders of the quadruplex engine.

Steam pressure 85 lbs. per gauge; water load 75 lbs. pressure; engine making 12 expansions of the steam and giving a mean effective net pressure of 26 lbs. on the piston; and pumping at the rate of 5,000,000 gallons per day through 1,000 ft. of 16-inch main before the water gets to the distribution system. The horsepower would be 150 at a piston speed of 100 ft. per minute. Now, supposing that the demand quickly increased 25 per cent, making a pumpage rate of 6,250,000 gallons per day, then the horsepower would go up to 187.5 by the increased piston speed up to 125 ft. per minute with the same cut-off and pressure; or, the same indicator card with a higher speed. But there would be an increase of 2.3 per cent in the power necessary to cover the friction in the force main alone, from the increased flow of water, and this would call for a corresponding increase in the mean effective pressure to a little over 27 lbs. To meet this the ratio of expansion would have to go down to 10, which would require a letting out of the cut-off accordingly.

It becomes apparent at once that the regular fly-ball governor and drop cut-off would not answer, because the attempt to increase the speed of the engine by drawing away the water would at once result in shortening the cut-off, which would be exactly the opposite of what it wanted. On the other hand, should the water be checked in its flow, and the pressure tend

to go higher in the force main, the opposite effect would be the result, and the ratio of expansion would need to increase and the power lessen, just to the extent of the decrease in the friction of the flow. With the fly-ball governor and drop cut-off, the attempt to check the speed of the engine by raising the water pressure against it would result in extending the point of cut-off by the corresponding drop of the governor, which would again be just the opposite of what would be needed.

From this it will be seen that the regulation of a pumping engine on direct service or on any service, is something more of a problem than regulating a mill engine; and it also appears that limiting the top speed is about all that the fly-ball governor can legitimately do, so far as automatic regulation is concerned and independent of the water pressure.

Up to the time that the use of the quadruplex pumping engine was discontinued in 1882, and was succeeded by the Gaskill horizontal pumping engine, also built by the Holly Manufacturing Company, there had been erected in water works stations about 80 of these machines, varying in capacity from 1,000,000 to 6,000,000 U. S. gallons, and aggregating a daily capacity of about 300,000,000 gallons, the average size being about 3,500,000 gallons per 24 hours.

CHAPTER XI

THE GASKILL PUMPING ENGINE

The Gaskill Pumping Engine, which appeared in 1882, was designed by H. F. Gaskill, the engineer and superintendent of the Holly Manufacturing Company, at Lockport, N. Y. It followed the Holly quadruplex engine, and was brought out by that company to provide a machine of lower cost and as being better adapted to the growing demand for higher steam economy and larger pumping units already beginning to make themselves felt even at that time.

The Gaskill is a horizontal engine of the Woolf compound principle, and in the form in which it is built makes a distinct type of pumping machinery. It was the first high duty pumping engine built as a standard commercial machine; and while giving a fairly high steam economy, is extremely compact and convenient. The engine is shown in Fig. 26 in perspective, and in Fig. 27 in section.

Although its general features of design and construction have been retained, improvements in some of the details have been made from time to time, and it has fairly held its own in the water-works field where the sharpest kind of competition has always existed. The engine is of the horizontal, beam, crank and fly-wheel, compound condensing type; and has two identical sets of steam cylinders and double acting plunger pumps connected to a single main shaft with two cranks at an angle of 90 degrees, and one fly-wheel located at the middle of the length of the crank shaft. In fact, the machine is really two complete pumping engines, which, with the exception of both using the same crank shaft, can be operated separately and apart from each other; either straight condensing or com-

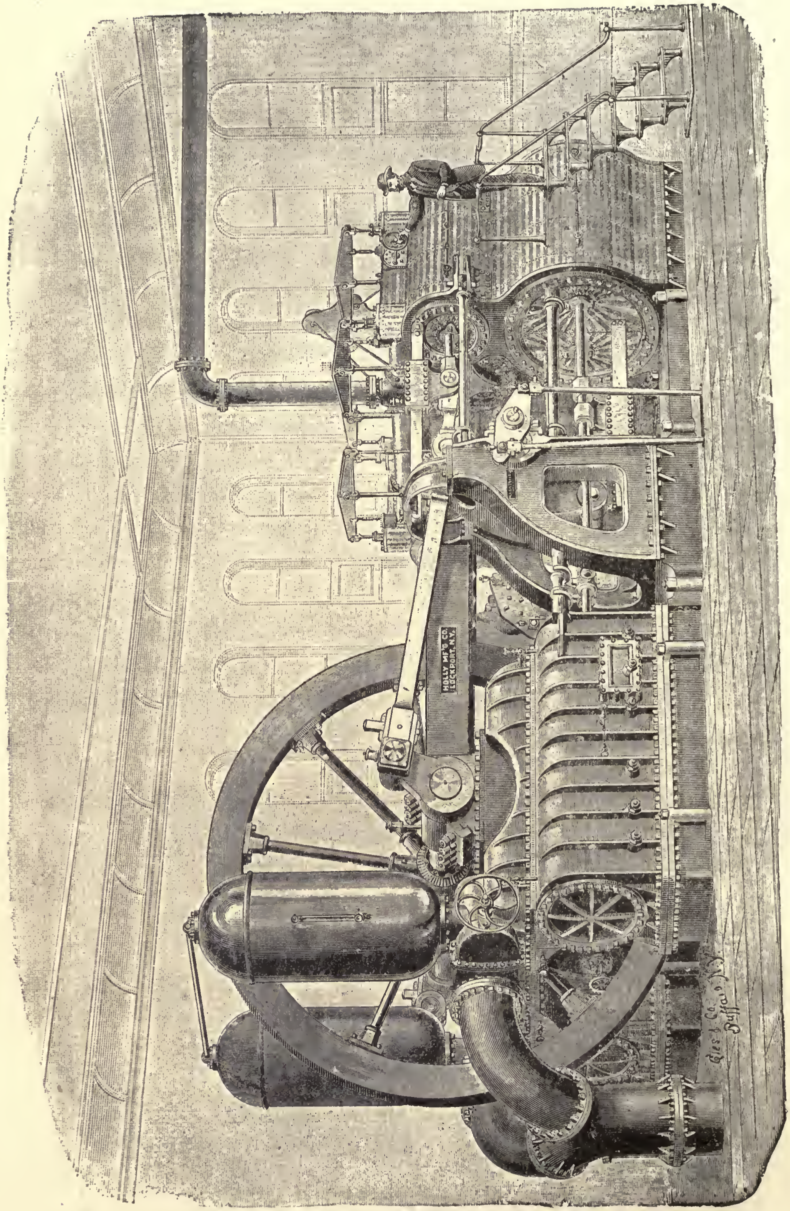


Fig. 26.—Gaskill Horizontal Compound Pumping Engine, Perspective.

HARVEY F. GASKILL.

pound condensing, and as lately produced, triple expansion as shown in Fig. 28. The two engines are located close together and side by side, with the center lines of all steam cylinders and water cylinders parallel. In the compound, which is the original class and by far the most built, there are two high pressure steam cylinders, two low pressure steam cylinders; and two water cylinders containing double acting plungers. The high pressure cylinders are mounted upon the low pressure, and the ratio of the high and low is generally 4 to 1 in area, or 2 to 1 in diameter. The two steam pistons of each independent engine move in opposite directions, and have their own cross heads connected by suitable links to opposite ends of a vertical rocking beam.

The two water cylinders are located in front of and in ine with the low pressure steam cylinders, the plunger rods and the low pressure piston rods being keyed to the same cross heads, these cross heads being connected to the lower ends of the rocking beams. About two thirds of the distance from the low pressure steam cylinders to the water cylinders there are located the heavy pedestals for supporting the center bearings of the rocking beams, and the general construction is such that the engine is self-contained in having the working strains within the lines of resistance. The pump plungers are arranged to work through solid rings, or through stuffing boxes as the case may be, according to the character of the water; these rings or stuffing boxes are bolted to an inner rib within the water cylinder usual in this type of water end. The water valves are small (see Fig. 29), and there are many of them, controlled by brass springs, and work upon brass seats screwed into the horizontal, flat valve decks, the suction deck immediately below and the discharge deck immediately above the plunger barrel. The water cylinders were rather square and box-like in their original form, and were thoroughly and heavily ribbed on the outside so as to combine the greatest strength with a good distribution of the metal; but the later engines have their water cylinders formed much more on the circular

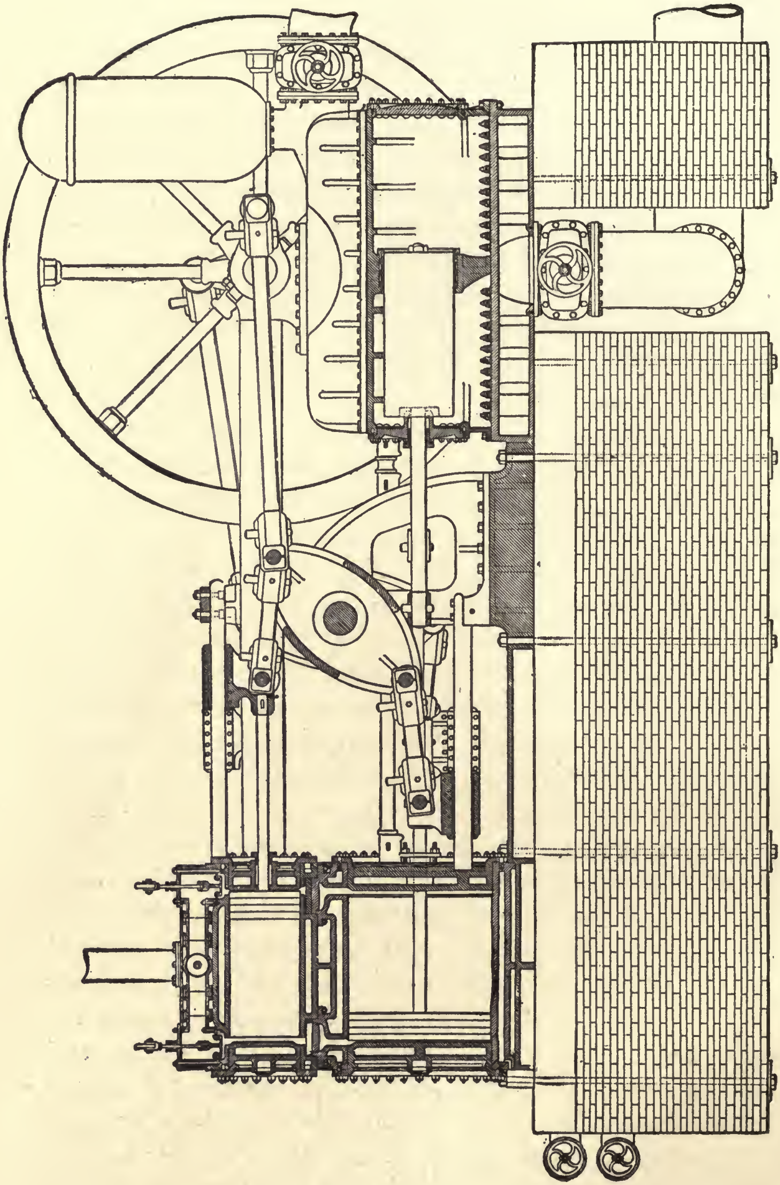


Fig. 27.—Gaskill Horizontal Compound Pumping Engine, Sectional.

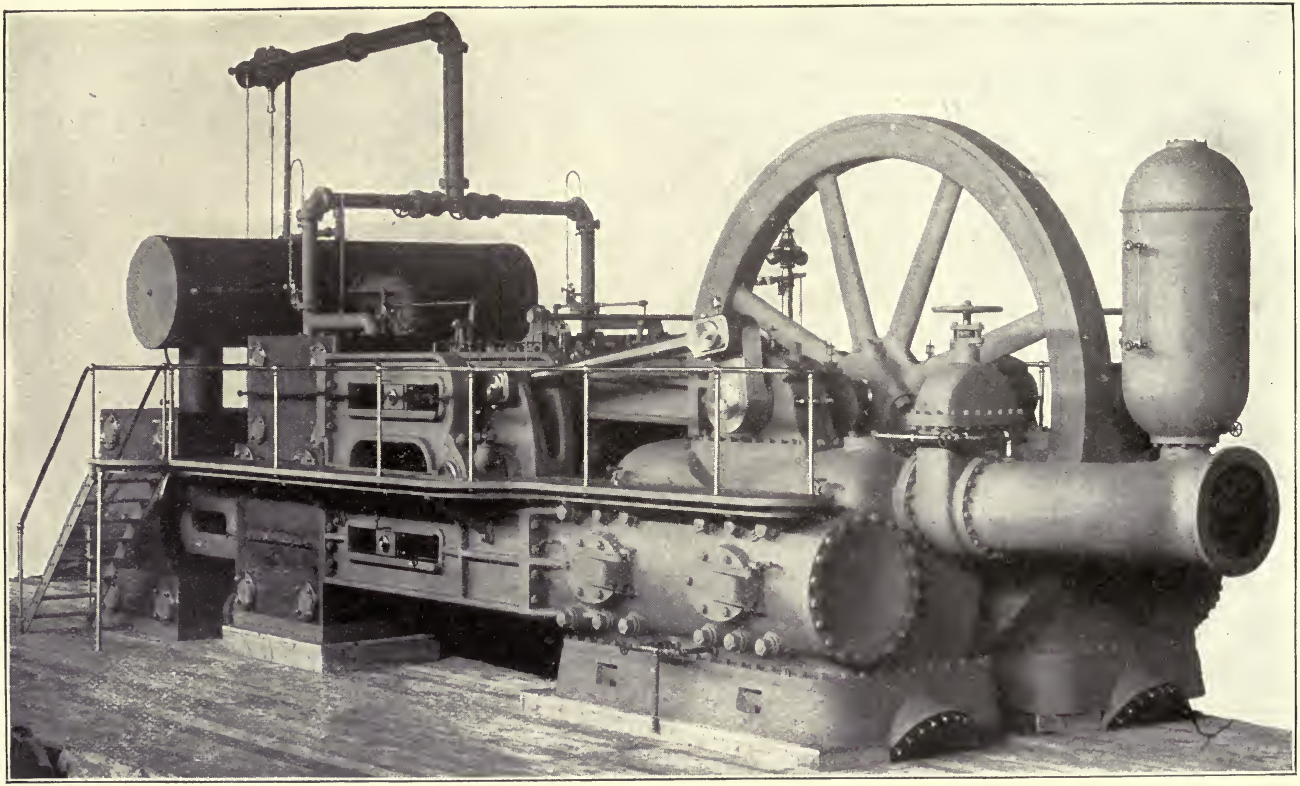


Fig. 28. — Gaskill Horizontal Triple Pumping Engine.

idea, and are undoubtedly considerably stronger than the original shape.

Above and below the water cylinders proper are situated the force and suction chambers, the suction chambers generally being connected by a cross pipe underneath, and at the middle of the length of the pump, by two 90 degree bends of easy radius, joining in a three-way casting from which the main suction pipe extends out at the extreme water end of the machine, and so out of the building or into a well within the building as the case may be. In the original engine the force chambers were bolted on top of the water cylinders, but in the later machines cast solid with the cylinders; on top of the force chambers are located the main pillow blocks for the crank shaft bearings, and the shaft extending across the engine carries one fly-wheel at the middle of its length; the cranks are secured to the ends of the shaft at 90 degrees from each other and in line for connection with the top end of the rocking beams.

In the early engines there was a bedplate extending completely under each half of the machine, and upon which the various members were mounted; but in the later engines the steam and water ends practically rest upon separate foundation piers, and are connected by means of heavy cast iron frames bolted in between the inboard ends of the pumps and the steam cylinders so as to hold them rigidly in place, and there are also cast iron girders extending from the inner frames to the main pillow blocks. The frames also contain the bearings of the rocking beams, and the cross head guides.

The discharge openings are located at the outer ends of the force chambers, and are connected across by means of 90 degree bends with their outer ends swung downwards at an angle of 45 degrees, until they meet in a three-way casting, from the lower opening of which extends the delivery pipe straight down through the floor, and thence carried to the most suitable point for the delivery of the water.

The Woolf compound steam end of the engine has steam jacketed cylinders; the early engines were fitted with poppet

induction valves on the top of the high pressure cylinders, and with gridiron exhaust valves which also serve as induction valves for the low pressure cylinder, the location of the cylinders permitting of this arrangement, which was admirably carried out in the design. The low pressure exhaust valves are also of the gridiron type, and by a short exhaust pipe sends the steam directly into the condenser. The regulation of the point of cut-off was accomplished by means of a rocking shaft with very limited motion controlled in its angular position by the pressure regulator already mentioned in connection with the quadruplex engine built by the same company. The poppet valves are operated by two revolving shafts extending lengthwise of the engine and driven by bevel gears directly from the main or crank shaft; the valve is moved by a vertical stem extending upward from the valve to the end of a short beam pivoted to a rigid column attached to the framing of the engine; the free end of the beam is connected to a link which extends downwards to one side of the strap of a small eccentric on the longitudinal shaft already mentioned. At the opposite side of the strap is a tailpiece shod with hardened steel, and as the shaft revolves, the tailpiece coming in contact with an arm on the rocking shaft also shod with steel, is prevented from further downward motion; the opposite side of the eccentric strap continues in its travel and so depresses the outer end of the beam, raising the inner end

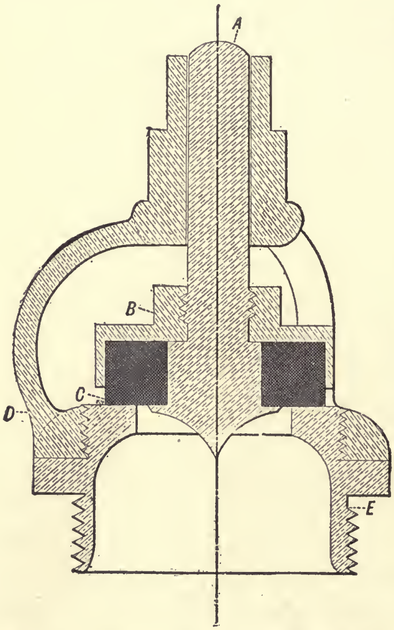


Fig. 29. — Pump Valves of Gaskill Engine.

of the beam, and opening the poppet valve. When the tailpiece is made to slip off from the rocking arm by the further movement of the eccentric, the valve quickly closes and cuts off the steam. The gridiron valves are driven by eccentrics from the same shaft that operates the poppet valves, one eccentric operating both high and low pressure exhaust valves.

The later engines are fitted with the Corliss type of steam valves and valve gear, operated by eccentrics on the main shaft, excepting that the high pressure induction valves, instead of having the detaching drop cut-off, are arranged so that the closure is made by positive means absolutely controlled by the engine. Fig. 30 and Fig. 31 show the general construction of the later Gaskill engines, although in some sizes the main cross head carries the inner end of the connecting rod, and the rocking beam is driven by a pair of links between which the connecting rod passes, whereas in the figures here given as in the original design the inner end of the connecting rod is carried by the top end of the rocking beam.

There are two air pumps, one for each side of the machine, driven from arms attached to the center shafts of the rocking beams, and are located in the space between the steam and water ends. The condenser, generally of the jet type, is located below and between the low pressure cylinders, giving short, easy, and quick passage for the exhaust steam.

The steam pistons are usually of the plain cast iron ring variety, with internal springs, although sometimes sectional packing is used. The water plungers are hollow and closed at the ends, with the rod extending clear through the entire length of the plunger, secured by a stoppered nut at the outer end. The plungers are arranged to work through different forms of rings according to the water, sometimes a plain solid ring, but mostly through stuffing boxes, as shown in the sectional elevation.

The operation of the engine is as follows:

Steam is admitted through the automatic cut-off poppet valves into the high pressure cylinders, forcing the piston for-

ward under nearly full boiler pressure until the point of cut-off is reached. The induction or admission valve then suddenly closes, and the remaining portion of the stroke is completed by the expansive force of the steam. When the high pressure piston has nearly reached the end of its travel, the gridiron slide valve, which acts as an exhaust for the high pressure and an induction valve for the low pressure, opens and the steam in the high pressure cylinder passes into the low pressure cylinder and drives the low pressure piston the length of its stroke in the same direction the high pressure has just traveled, the steam expanding to four times its volume, at the time of the high pressure exhaust. The release from the low pressure cylinder is accomplished by means of the gridiron exhaust valve at the side of the cylinder; the steam, now expanded down to a very low point, escaping into the condenser where it is condensed again into water. This operation is repeated at every stroke of the engine by each high and low pressure steam cylinder alternately; the change from poppet and gridiron valves to Corliss valves does not change the principle of the machine in any particular; the cut-off and expansion in the high pressure, and the full stroke expansion in the low pressure remaining the same, as also does the combined exhaust and the induction valves between the two cylinders.

The Gaskill pumping engine is the only consistent attempt so far made to increase the economy of the original Worthington form of machine by adding a crank and fly-wheel to the four steam cylinders and two water cylinders. There is no doubt that it was brought out for this very purpose when the low duty non-rotative machine with its limited ratio of steam expansion was at the zenith of its time, but, as already pointed out, the Gaskill was soon followed by the Worthington high duty, and this brought the two rival establishments to nearly equal commercial terms, so nearly so in fact, that one of them a few years later succeeded in swallowing the other. But the non-rotative engine lost the claim for simplicity of construction so long and so strongly put forward, and justly too for so many

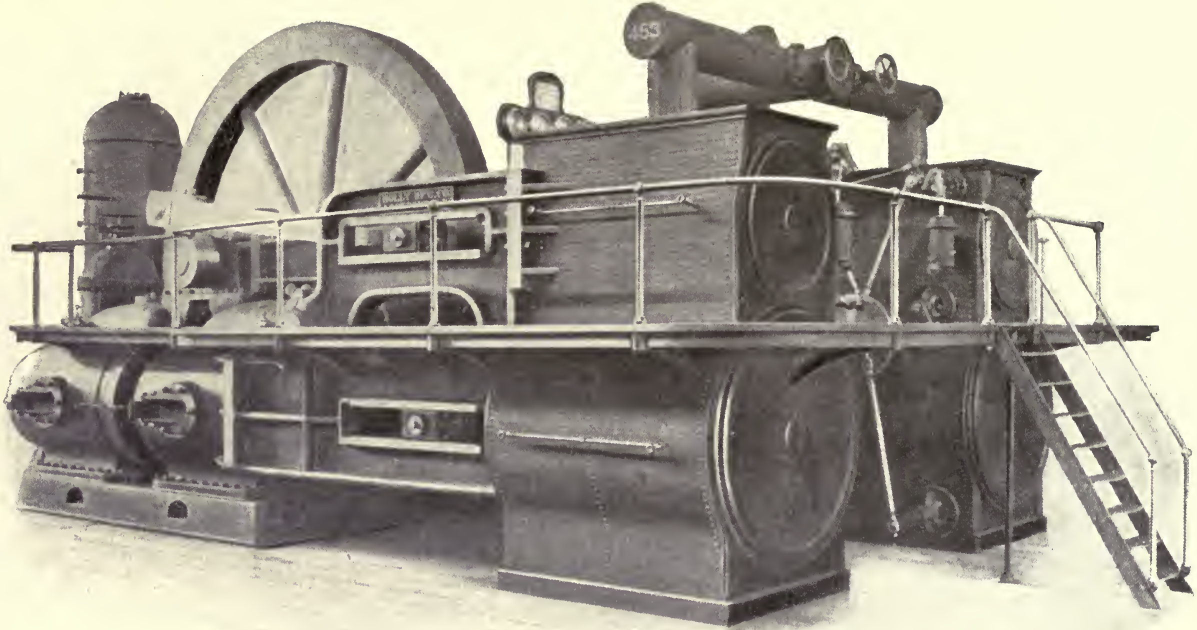


Fig. 30.—Latest Gaskill Horizontal Compound Pumping Engine.

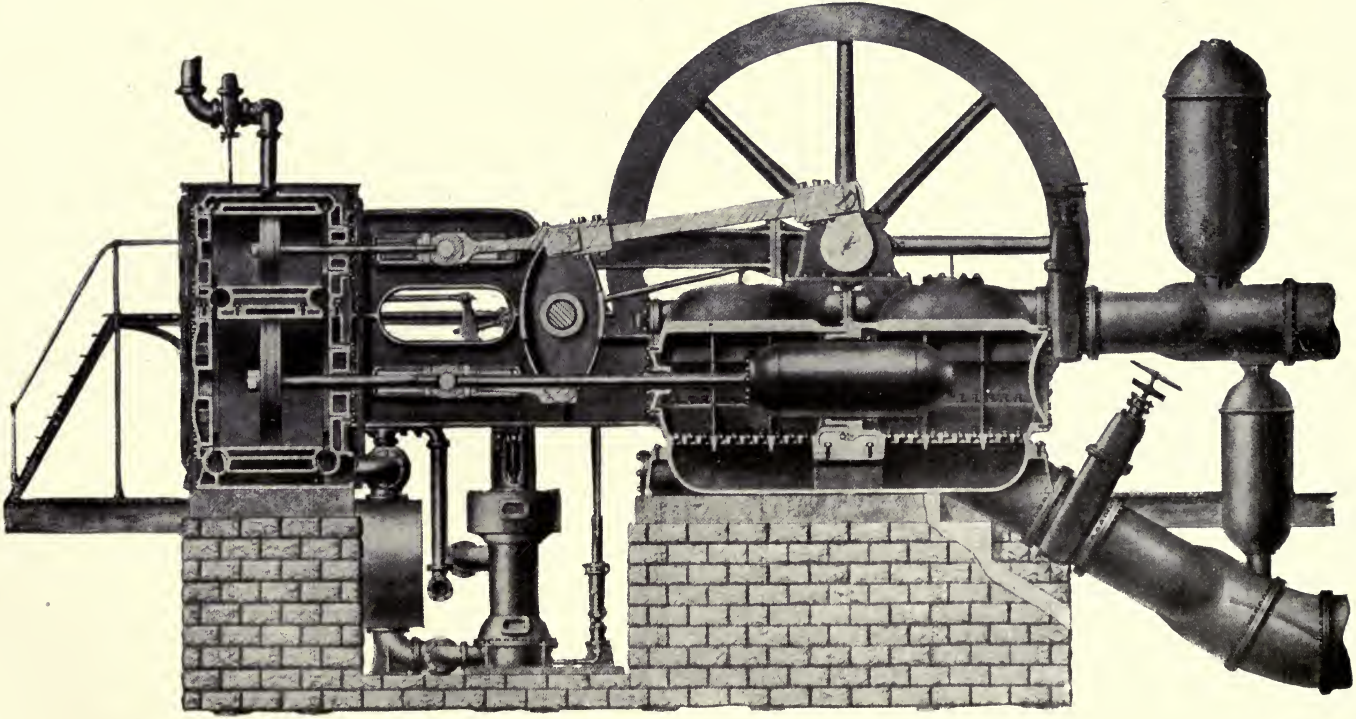


Fig. 31. — Latest Gaskill Horizontal Compound Pumping Engine.

years, by the direct acting advocates; which all goes to show that low duty means simplicity of machinery, and high duty means more complicated machinery; the main question being where to draw the line so as to balance the account to a reasonable extent. The original type of Worthington low duty engine, the high duty Worthington, and the original Gaskill engines have moving parts and joints as follows:

Worthington Low Duty.

Steam pistons	4
Steam valves	4
Water plungers	2
Stuffing boxes	14
Journals	28

Worthington High Duty.

Steam pistons	4
Steam valves	16
Water plungers	2
Stuffing boxes	35
Journals	90

Original Gaskill.

Steam pistons	4
Steam valves	12
Water plungers	2
Stuffing boxes	20
Journals	100

The Gaskill pumping engine completely supplanted the quadruplex machine, and was arranged to regulate by the same means as its predecessor, the poppet valves as employed in the new machine lending themselves very readily to the purpose. This engine requires practically no more room than its low duty rival, and in fact represents the direct acting machine with the high pressure cylinder placed on top of the low pressure instead of in front of it, and this shortened up the design so that the increased distance between the steam and water ends to accommodate the rocking beams resulted in about the same total length over all. The mounting of the fly-wheel

above and between the pumps only made a slight increase in the height without increase in the length of the water end, so that any building large enough for one gave sufficient space for the other.

Since the introduction of the Gaskill pumping engine in 1882, there have been furnished to various cities and towns about 200 of these engines, with an aggregate pumping capacity per 24 hours of approximately 1,250,000,000 U. S. gallons; the smallest built, of 1,000,000 and the largest 20,000,000 gallons daily capacity; the average capacity being about 6,000,000 gallons.

CHAPTER XII

THE REYNOLDS TRIPLE EXPANSION PUMPING ENGINE

The vertical, triple expansion, crank and fly-wheel condensing pumping engine, which about twenty years ago, 1886, developed into a pronounced type, has since been repeated many times, and has been copied in all essential features by most of the builders of large sized pumping engines in the country. It was originally designed at the works of the Edward P. Allis Company of Milwaukee, Wis., by Irving H. Reynolds, under the supervision of Edwin Reynolds, the general superintendent of the establishment; this triple expansion machine, which is of 6,000,000 U. S. gallons daily capacity, being a very natural development of the three crank compound pumping engines designed by Edwin Reynolds for the Allegheny, Pa., water works in 1883, and who had just previously, in 1881, designed and built his first large pumping engine for the Milwaukee water works; this first machine being a peculiar form of beam engine, compound condensing, with one bucket and plunger pump directly beneath the steam cylinders; and was erected in the North Point pumping station in 1881. (See Fig. 32 for a general view, and Fig. 33 for a sectional elevation.) This form of engine was never repeated, probably on account of cost, although it gave a little over 104,000,000 duty per 100 lbs. of coal burned on the grates.

The next step after the Milwaukee beam engine of 1881, which was of 12,000,000 U. S. gallons capacity, was the compound condensing engines of 6,000,000 gallons capacity, two of which were built and installed for Allegheny, Pa., in 1883 and 1884, where they were tested by Professor David M.

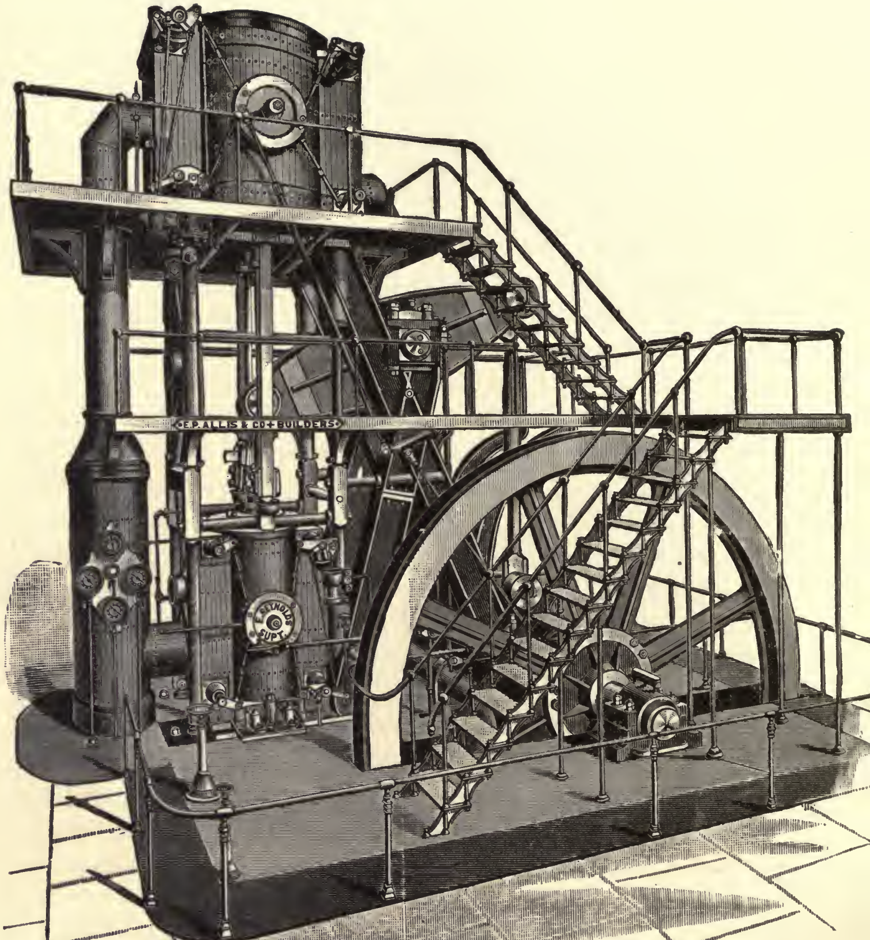
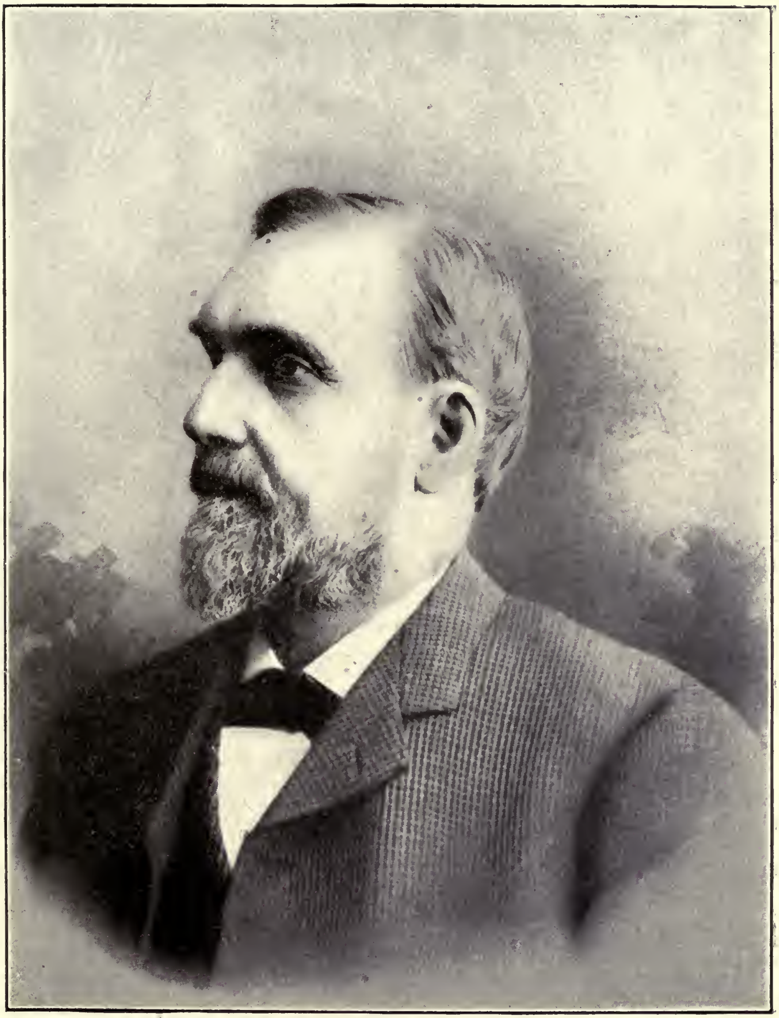


Fig. 32.—Reynolds Beam Pumping Engine, Milwaukee.



EDWIN REYNOLDS.

Green and the writer in the latter year; the duty obtained was 107,000,000 ft. lbs. per 1,000 lbs. of steam, or, as it was then put, per 100 lbs. of coal based upon an evaporation in the boilers of 10 to 1.

In the Allegheny engines the departure was radically made

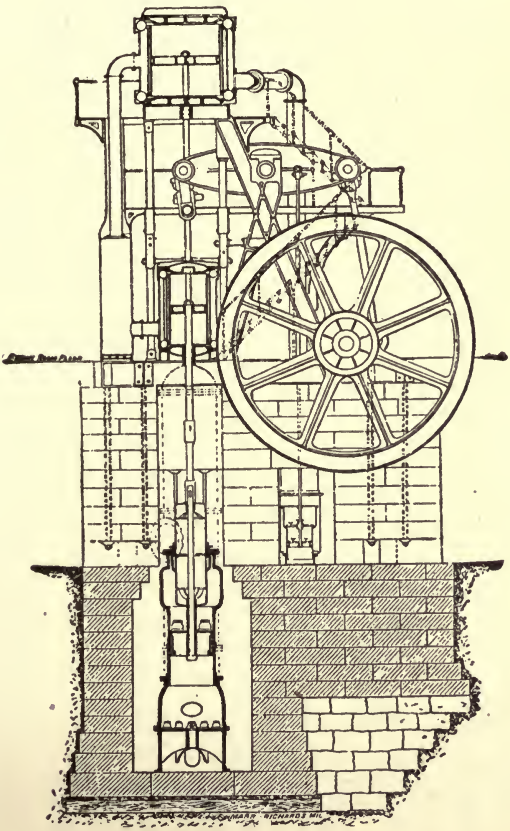


Fig. 33. — Sectional View of Reynolds Beam Pumping Engine.

from four displacements per revolutions, which had been the usual delivery method, to three displacements with the cranks at angles of 120 degrees around the circle; and it immediately cut down the number of steam cylinders from four, as in the

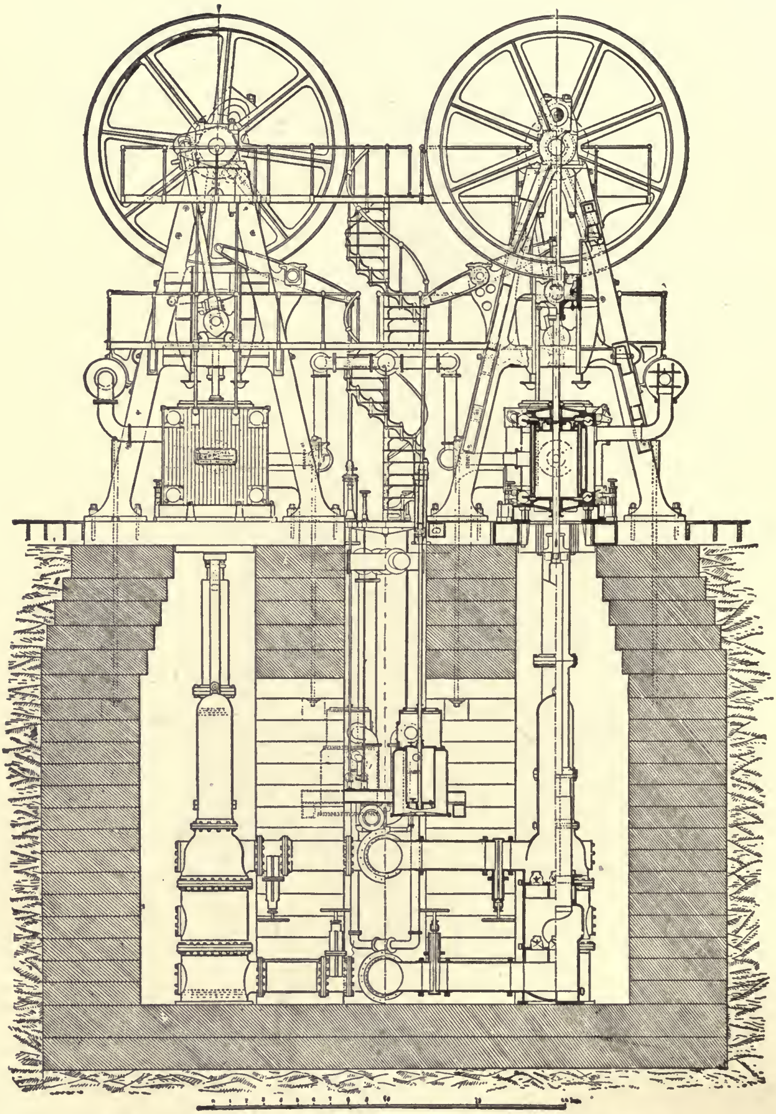


Fig. 34.—Reynolds Three Cylinder Compound Pumping Engine.

Worthington and Gaskill engines, to three, and produced a practicable high type of pumping engine of great simplicity. See Fig. 34 and Fig. 35 for front and side elevations, sections, etc., showing one high pressure steam cylinder and two low pressure cylinders, with three single acting, outside packed plunger pumps. These engines were two in number, exactly alike, both in general arrangements and in detail, of the vertical, three cylinder, compound rotative type, with large receivers between the high and low pressure cylinders. Each steam piston operated a single acting solid plunger pump, placed in a pit vertically underneath each steam cylinder. The center lines of the three steam cylinders and the three pumps are all in the same vertical plane in which above the steam cylinders is located the main crank shaft in two pieces, which carry two fly-wheels weighing about 10 tons each, one wheel over each of the two spaces between the three cylinders. The wheels are placed overhead, the crank shafts mounted in pillow blocks at the top of the A frames, secured to the main bedplates, the bedplates resting upon the foundations, and the steam cylinders secured to the bedplates between the A frames.

The valve gear is of the Corliss type with details designed by Edwin Reynolds, adjustable by hand on the two low pressure cylinders, while that of the high pressure is very effectually regulated by the ordinary fly-ball governor, the machinery being always operated at full speed capacity. The receiver is cylindrical, horizontal, and mounted behind the steam cylinders at about the level of the cylinder tops. The cylinders are steam jacketed, packed around their outside barrels with mineral wool, and lagged with black walnut; the receiver is also covered with non-conducting material and lagged to correspond with the steam cylinders. The water of condensation from the steam jackets and receiver was discharged through traps operating automatically, and the receiver has no reheating coils.

The pump valves were originally of the Cornish double beat type, of brass; there were seven suction and seven discharge

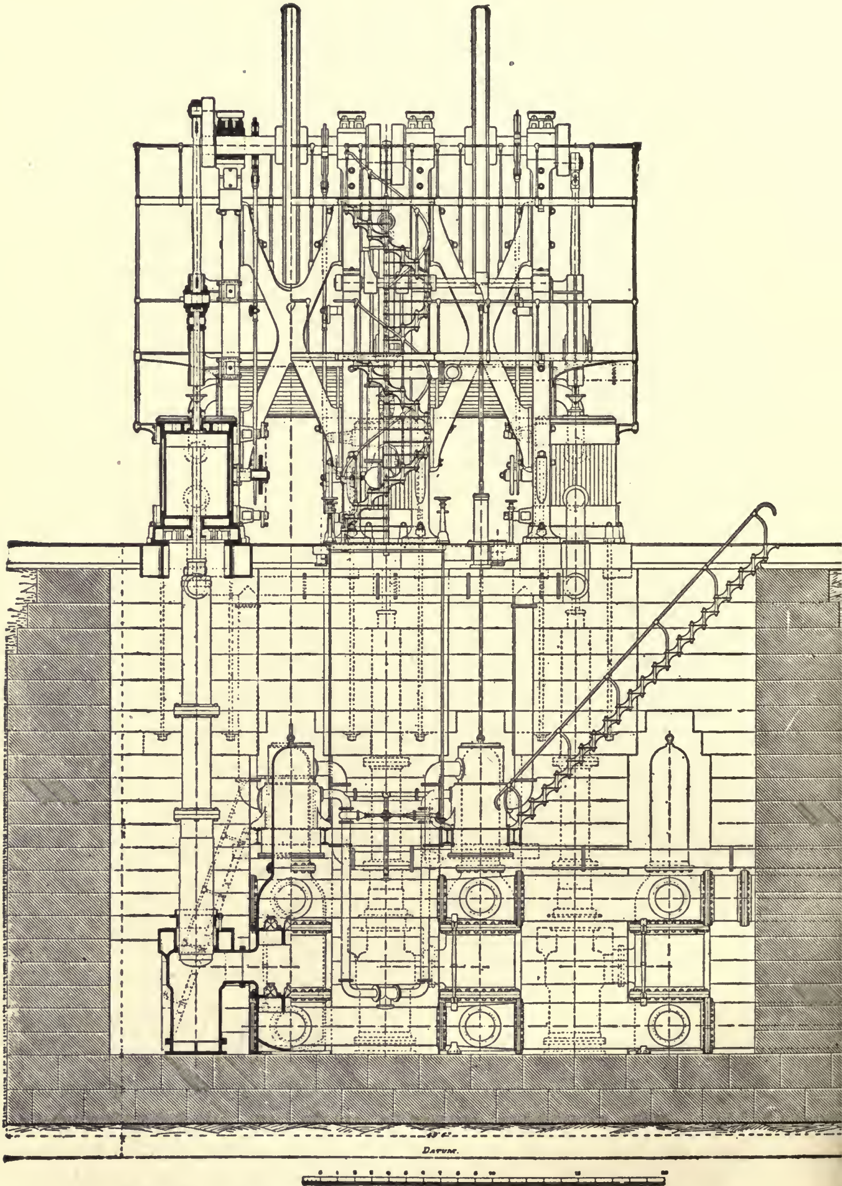


Fig. 35.—Reynolds Three Cylinder Compound Pumping Engine.

valves for each plunger, and right here comes in an illustration of how satisfactory permanent results are sometimes obtained in trying to escape from a difficulty which has presented itself. The waters of the Allegheny River contained sufficient troublesome floating refuse to interfere with the proper operation of the brass Cornish pump valves, when such refuse came between a hard non-elastic seat and valve, which of course a brass valve and seat would be. The writer, who was then in the employ of the Allis Company, watched the operation of the pumps and reported from time to time, until it was evident that the hard unyielding pump valves seating themselves upon equally hard brass seats, were decidedly objectionable under Allegheny River conditions. But there were the main pump chambers, representing a good deal of cost and permanently set in place in the pump pit. Each valve deck with seven holes, each 8 inches in diameter through it, entirely unfitted for the application of rubber valves in the usual way, and in fact totally unable to furnish the proper valve area as generally applied. The remedy employed resulted in a form of cage construction shown in Fig. 36 and Fig. 37; these cages held in place by one large center bolt which took the place of the central bolt of the Cornish valve. This overcame the difficulty completely and conclusively, and in fact so well that it was adopted as a permanent detail of the Reynolds pumping engines and is used even to-day in the construction of this type of machinery. The illustrations given in the figures of this engine are from an old engraving, and the dome-like form of the old Cornish valves may be plainly seen in the picture, especially with the aid of a magnifying glass to enlarge the detail.

The next Reynolds pumping engine was a vertical two plunger compound condensing machine, having outside packed plungers, which type of plunger has always been favored by the designer of these engines. The general design of this two plunger engine, or as much of it as shows above the engine room floor, appears in Fig. 38 and is of the "see-saw" class,

where two plungers have directly opposite movements at the same time, making a good enough delivery, perhaps, for reservoir work with a direct force main, but very objectionable where consumers of water are attached to the main, or distrib-

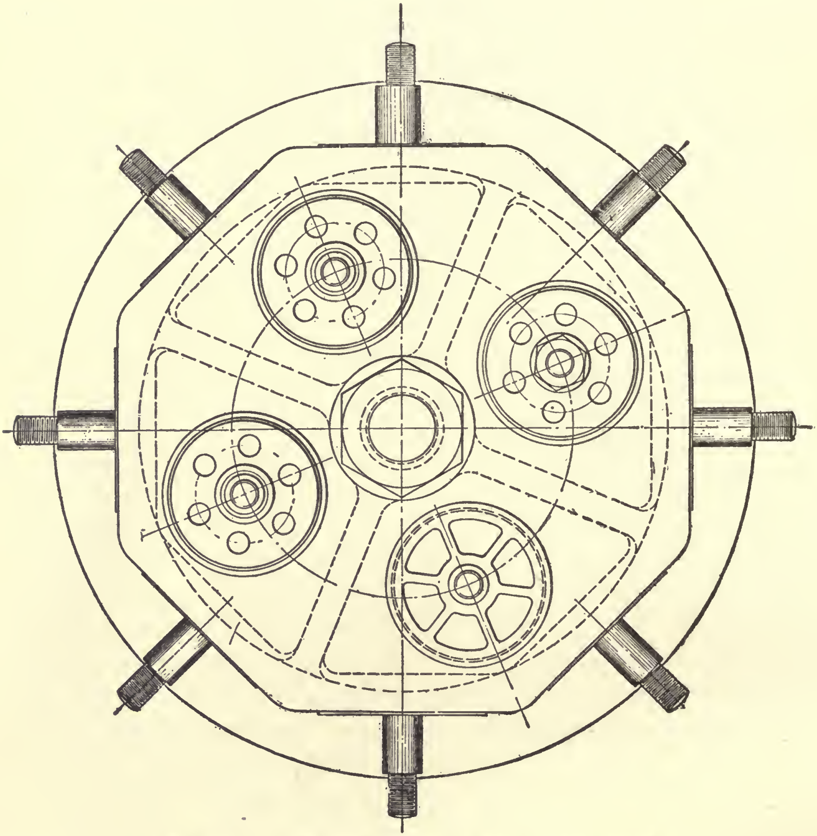


Fig. 36. — Cage Construction of Pump Valve Seats.

uting pipes leading from a main connected with this type of two plunger machine. As an example of the possible and probable annoyance to consumers arising from the use of two plunger "see-saw" pumping engines, the writer recalls an extended examination into the matter in Milwaukee, and

could count every stroke of the pumps at the kitchen faucets of every house within the high service district supplied by one of these engines.

Pumping engines of this vertical two plunger compound condensing type, were designed and built for Milwaukee, St. Paul, and some other cities, and the final one for Hannibal, Mo., mention of which has already been made. In the latter engine

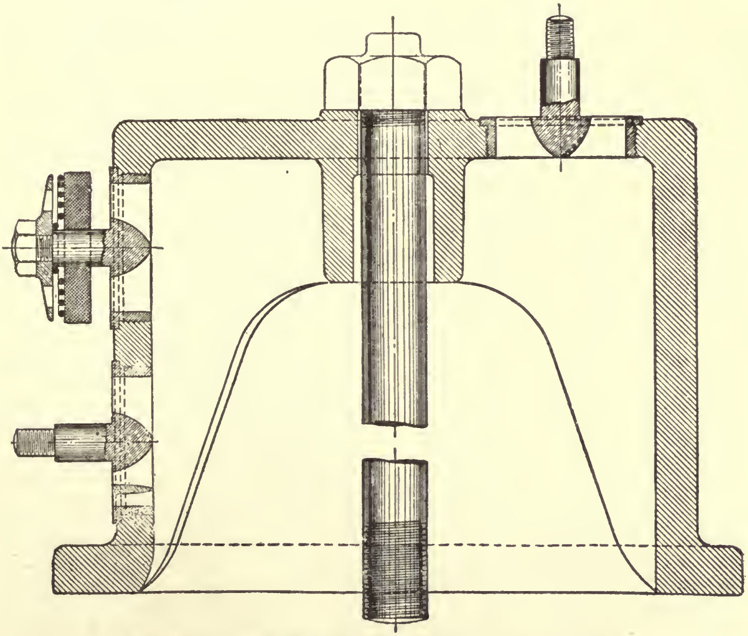


Fig. 37.—Cage Construction of Pump Valve Seats.

some of the important features seen in the later triple expansion design first appeared; such as closer clearance and smaller waste room at the steam cylinder ends, especially in the low pressure, the last cylinder to receive the steam before it is lost in the condenser. This is the first pumping engine of the Corliss type steam end with the valves across the cylinder heads, with the single exception of the Pawtucket engine designed and built by Geo. H. Corliss in 1877, although this

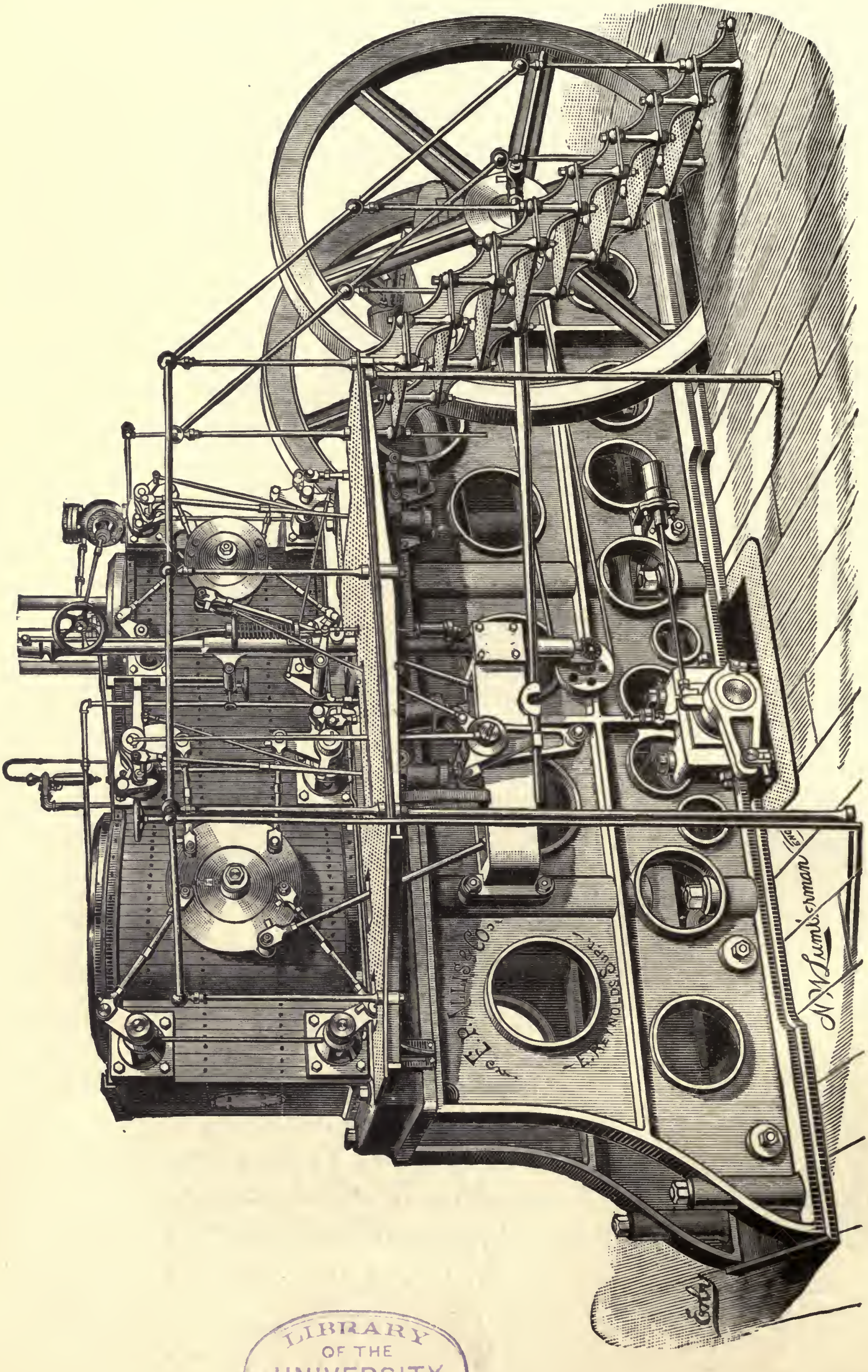


Fig. 38. — Steam End of Reynolds St. Paul Pumping Engine.

location of steam valves was not new even at that time, as the Corliss Centennial power engine at Philadelphia in 1876 was so provided. This detail in engine design is an extremely important one where high steam economy is sought, because the less the waste room which has to be filled with steam at the cylinder ends at each stroke, the nearer the approach to theoretical cylinder displacement. And the advantage of the cross-the-head form of valves is that they avoid the corners at the ends of the steam and exhaust ports formed by the difference between the straight line of the valve seat and the circular line of the cylinder bore. The introduction of poppet valves into this type of pumping engine has still further reduced the waste room, and brought it down to merely the necessary clearance for the piston and head, the matter of port room having been entirely wiped out of the case with the poppet form of admission and exhaust valves.

Immediately after the Hannibal engine had given a duty of 118,000,000 ft. lbs. per 1,000 lbs. of steam, the writer made a duty test of the same type at the high service pumping station of the Milwaukee water works on Grand Avenue, but the latter engine had the old-fashioned cylinders with the valves at the corners, and failed to reach 95,000,000 ft. lbs. duty per 100 lbs. of coal burned, but making 99,000,000 duty per 1,000 lbs. of steam. About a month later a test of a similar engine at the St. Paul water works of fairly good size, demonstrated beyond all doubt in the light of the Hannibal experience that the small clearance, the only difference in these engines, was the key to a good deal of improvement in steam economy, and as it cost no more to build the engines this way, of course it was the proper line to follow.

After further discussion regarding the Milwaukee compound engine, which was not up to contract duty requirements, the suggestion naturally enough came about of its replacement by the then new idea of a triple expansion machine, with the steam cylinders fitted with the valves across the heads; and so the agreement upon the new type, really the joint efforts of

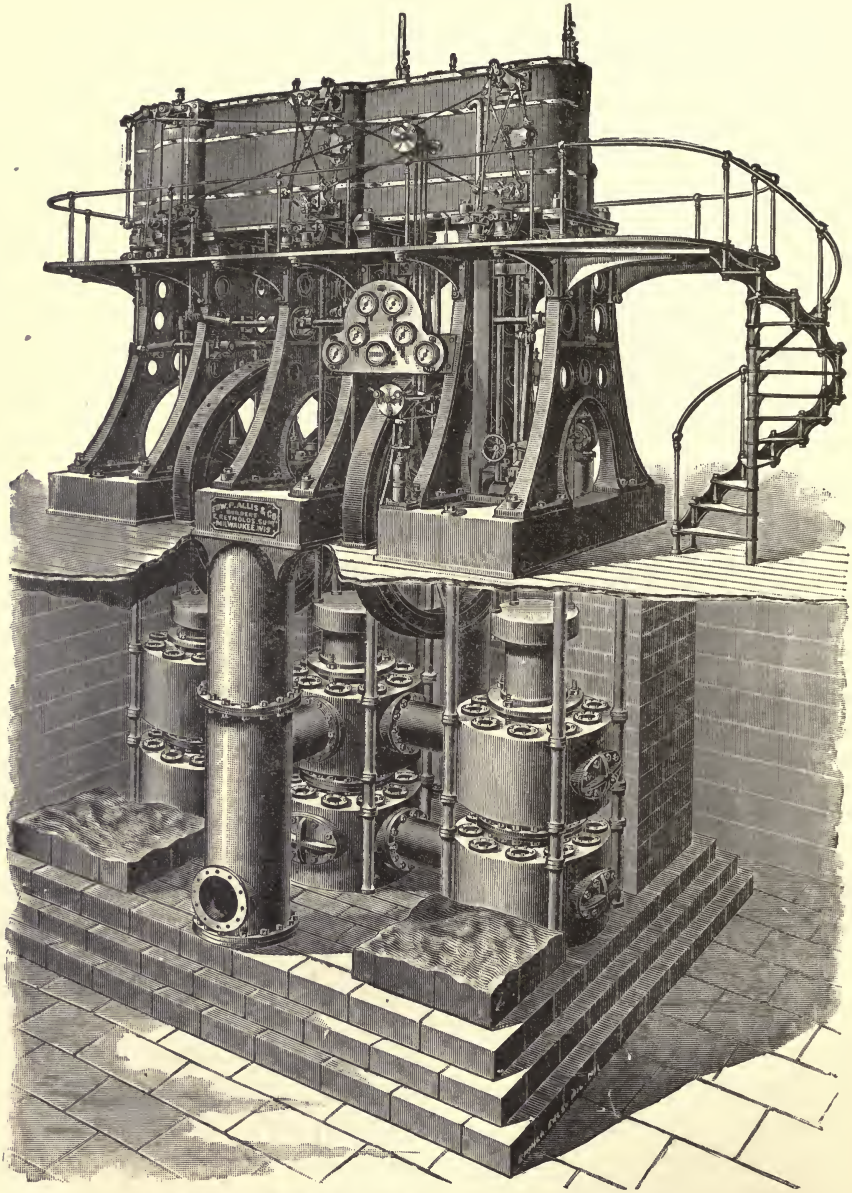


Fig. 39. — Reynolds Triple Expansion Pumping Engine.

three people, was arranged with the city. This is the first triple expansion pumping engine in the world's record, and it proved an immediate success. A general view of this machine is given in Fig. 39, and it will be observed that it looks very much like the latest productions of this type, the principal features being unchanged at the present time, although some refinements in the smaller details have been carried out. The duty of this newcomer in 1886 ran up to about 122,452,724 ft. lbs. per 100 lbs. of coal, with return tubular boilers carrying 80 lbs. gauge steam pressure; and its successors from the same establishment have ever since held the high duty record, the present figures reaching in April, 1906, a little upward of 181,000,000, which from all evidence, investigation, and probabilities, is very close to the limit of the accomplishment in the high duty line of effort, with the use of dry saturated steam, and for 1,000 lbs. consumed, including jackets and reheaters in the receivers.

This original triple, of 1886, has the following dimensions:

High pressure cylinder, 21 inches diameter.
Intermediate cylinder, 36 inches diameter.
Low pressure cylinder, 51 inches diameter.
Pump plungers, three single acting, $23\frac{1}{2}$ inches diameter.
Stroke of all, 36 inches.

The steam pressure specified by the city was not to exceed 80 lbs. per gauge, and this rather low pressure was stipulated because of the boilers already in the station and which the city did not desire to replace at that time. The total load on the plungers was about 48 lbs. to the square inch, and the capacity of the engine was 6,000,000 gallons per 24 hours. There were plenty of predictions made that with the low steam pressure of 80 lbs., the triple expansion idea would prove unsuccessful so far as economy of its steam was concerned; but there are records of duty of 129,000,000 per 1,000 lbs. of steam from this engine, which calls for a little less than $9\frac{1}{2}$ lbs. evaporation, a very reasonable figure with the clean anthracite

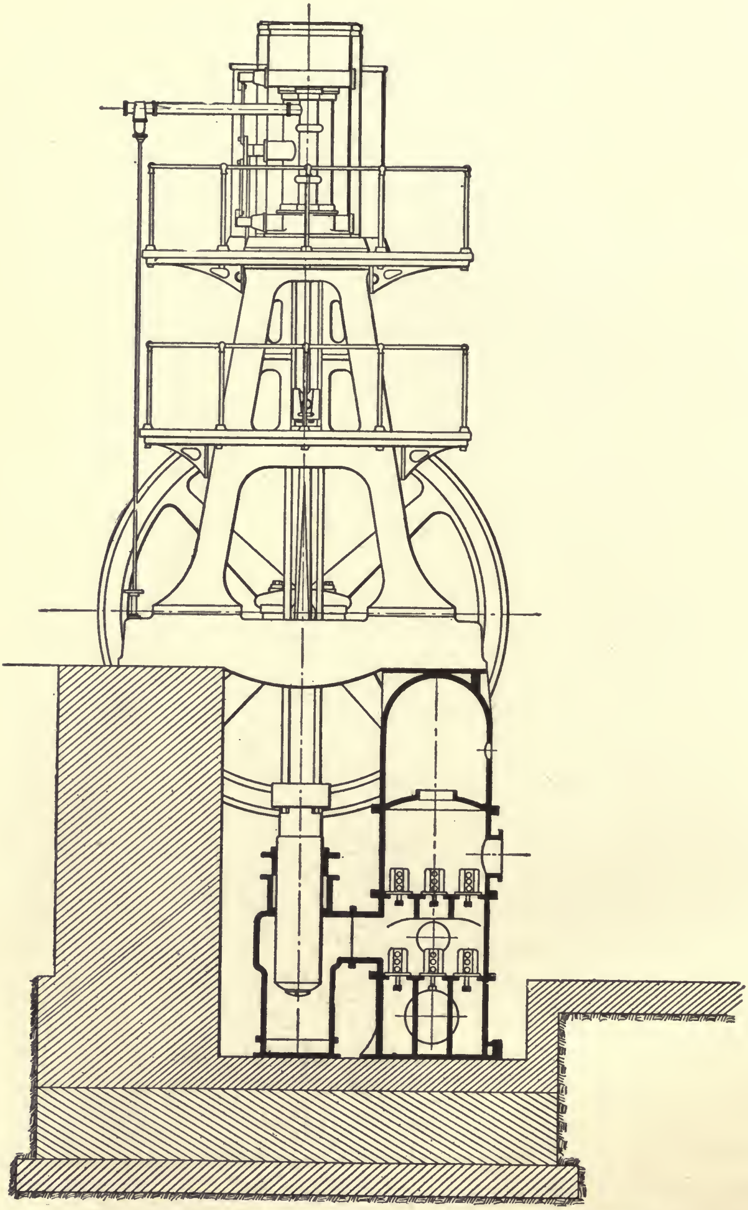


Fig. 40.—North Point, Milwaukee, Triple Pumping Engine.

coal used, the boilers in good order, and with the moderate steam pressure to evaporate the water against; that is to say, a coal duty of 122,452,724 and a 1,000 lbs. of steam duty of 129,000,000 call for 9½ lbs. evaporation in the boilers under 80 lbs. gauge steam pressure.

These results bear out the idea held for years by the writer, that if the gain from multiple expansion is due to a considerable reduction in the range of temperature within the cylinders, then the benefit should show with moderate pressures as well as with high pressures. Of course the higher pressure of initial steam, with the same terminal pressure, will give a greater economy; but even at that, the steam must be protected by reducing the range of temperature, and as 4 or 5 expansions is about the limit in one cylinder, there should be and is a very considerable improvement in steam economy in tripling, even with 80 lbs. pressure of steam. At all events, the steam pressure has been more than doubled in 20 years, and the duty only increased 40 per cent clear up to the new record of 181,000,000 now held at St. Louis, Mo., by a successor of the original triple engine from the Milwaukee concern.

This form of construction of the Reynolds pumping engine marked and identified this type, which made another decided record stride forward in the large North Point engine with a 24 hour capacity of 18,000,000 U. S. gallons, at the main station of the Milwaukee water works in 1892, with a duty of 154,048,704 ft. lbs. per 1,000 lbs. of dry steam. (See Fig. 40.)

Still another step forward was taken with Engine No. 10 in the St. Louis water works in 1900 with a 24 hour capacity of 15,000,000 U. S. gallons, which developed a duty of 179,454,250 ft. lbs. per 1,000 lbs. of dry steam. The record has again been broken, in April, 1906, also at St. Louis, with one of the same type and make of machinery, when a duty of 181,068,605 ft. lbs. per 1,000 lbs. of dry steam was obtained.

This type of pumping engine, the general appearance of which is given in Fig. 41, as its steam or power end appears above and below the floor of the engine room, is really three separate vertical

steam engines set side by side. Each is upon its own bed-plate and with its own framing; the first or high pressure cylinder takes its steam directly from the boiler, and as nearly to boiler pressure as it is possible and practicable to get it, and exhausts into a receiver which acts as a sort of boiler for the second or intermediate cylinder; this second or intermediate cylinder taking its steam from this first receiver and exhausting it into a second receiver which answers for the boiler for the third, last, or low pressure cylinder; and the third or low pressure cylinder taking its steam from the second receiver exhausts into the condenser. The idea is to keep the pressure in the two receivers exactly at the terminal pressure of the cylinder from which the steam comes, so that there will be no drop of pressure when the exhaust takes place, the receiver being made large enough to prevent any appreciable change in its pressure from the incoming and outgoing steam. The result of this operation is to get the benefit in work of all of the expansion, and to allow no expansion to take place without a corresponding effect upon the pistons of the three cylinders. It is just the same as making one large indicator card in one cylinder, and then cutting it by horizontal lines into three parts, which give equal power in each diagram. This is not exactly and precisely done in practice, but it is approached very closely, and it is the method for comparing the effectiveness of the steam distribution.

The exhaust steam from the high and intermediate pressure cylinders is "reheated" by means of steam circulating in coils within the bodies of the receivers; and the cylinders are jacketed by steam, both at the sides and heads, in the most effective engines. One of the most efficient arrangements of steam jacket and receiver heating pipes, perhaps as good as any, is as follows:

Boiler pressure, or as nearly as may be, from the main steam pipe near the engine, passes to the top of the high pressure jacket, and out at the bottom of the jacket. Then through a branch in the high pressure jacket outlet to a Flynn trap,

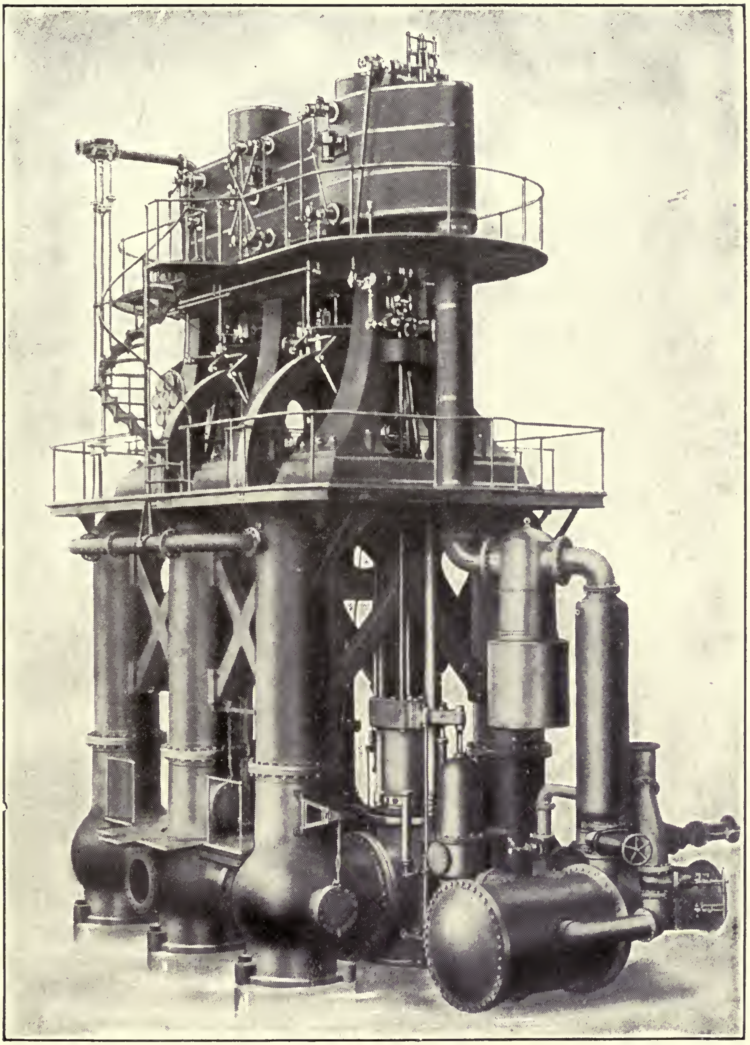


Fig. 41.—Reynolds Self-Contained Triple Pumping Engine.

and from the discharge of this trap to the top of the low pressure cylinder jacket. Another branch from the high pressure jacket outlet goes to the coil in the first receiver at the top; then out of the bottom end of this first receiver coil and continues to the top of the intermediate pressure jacket, then from the bottom of the intermediate jacket to the top of the low pressure jacket. The steam between the high pressure jacket and the first receiver coil is throttled so as to reduce the pressure in the latter, the natural condensation for the high pressure jacket going through the Flynn trap already mentioned, to the low pressure cylinder jacket. The condensation from the working steam in the first receiver goes to the top of the coil in the second receiver, and from the bottom of the second receiver coil to the top of the low pressure jacket. No reducing valves are used, but the pressure from high to low is regulated by two globe valves in the pipe, one valve directly after the other. The final outlet from the low pressure jacket at the bottom goes to a water seal in the basement of the building, the pressure being so low that no steam trap is required.

High pressure jacket has boiler pressure.
Intermediate jacket has 40 lbs. steam pressure.
Low pressure jacket has 0 on the ordinary steam gauge.

The water end of the Reynolds engine consists of three pumps, one of the pumps below and in line with each steam cylinder. The main framing is very heavy in comparison to the work to be done by the engine and, although apparently massive, is in good proportion; but of course beyond a reasonable calculation as to factors of safety the real needed strength of machine framing is not known excepting by experience. The designer of this machinery was once asked by a party inspecting one of these engines if the framing was not too strong, and the reply was "If it is too strong nobody knows it, but if it were too weak everybody would know it."

In some forms of the engine the framing is shaped like a double letter A, while in late machines what is known as a

single A frame is used. At the top of the A frames are placed the steam cylinders, and the bottom or feet of the framing rest upon and are securely bolted to the main bedplates in which are located the main pillow blocks. There are four pillow blocks, one at the inner edge of the high and the low pressure bedplates, and two on the intermediate bedplate. These four pillow blocks support two main or crank shafts which carry the fly wheels, the wheels, two in number, swinging in the two spaces formed by the three bedplates. The fly wheels are rather heavy, sometimes weighing in the larger engines as high as 30 tons each. These wheels may seem heavy, but the dangerous effects of fly wheels on water columns and force mains has been very much overstated by advocates of the non-rotative forms of pumping engines. The fact is that, with the low rate of revolution of a proper crank and fly wheel pumping engine, the drag produced by such a stubborn load as water largely dominates and controls the machine, and the writer was enabled several years ago to try some experiments upon this point which clearly illustrates how little real effect the fly wheel exerts beyond the regulation of the motion of the engine with reference to the cutting off and expansion of the steam.

The case was as follows: a new pumping engine of this type and of large size had been started but a short time, had been running a very few days, and naturally was not yet in perfect adjustment. Occasionally it was observed to come very quietly to a complete stop, and after considerable investigation it was noted that one of the cut-off hooks did not always get hold of the steam arm catch, on account of the dash pot rod being too short and the catch going down too far now and again. The length of the rod was properly adjusted and then all went well, but the suggestion arose to slip a thin wooden wedge between the hook and catch after full speed had been reached by the engine, to see the effect on the machine and what the fly wheels would do in the way of carrying the engine along against the water column, the working head being about 220 ft.

The result was rather surprising, for even with a three cylinder engine having cranks at three points on the circle 120 degrees apart, taking steam six times per revolution, there was not enough momentum in the fly wheels and moving parts to carry the engine one revolution and have the single cut-off hook operate after having been interfered with for a single stroke.

As already mentioned, there are two shafts in the Reynolds pumping engine, extending from high pressure and from low pressure bedplates to the intermediate bedplate, and at each end of each shaft is secured a crank, the two middle cranks coming nearly together below the intermediate cylinder, being considerably larger and stronger than the outer cranks below the high and low pressure cylinders. The connecting rods extending from the cross heads to the crank pins are of liberal length, and, as the cranks are set at 120 degrees from each other, the motion of each pump plunger is controlled by its own crank, and so the plungers acting in conjunction produce an extremely even and satisfactory flow of water. The pumps being single acting deliver water only on the downward stroke, but the plungers are weighted to half the difference between the suction and delivery loads so that the steam cylinders have equal work on both upward and downward strokes, and the only work going through the crank shafts is that which is given to and given back by the fly wheels on account of the expanding steam.

The main pumps, three in number, are of the single acting, outside packed plunger pattern, secured beneath the bedplates directly in line with the steam cylinders so that the force and resistance of the pumping are in the same center line, thus making the machine self containing with reference to the work it is doing. The pump barrels proper, or the plunger barrels as they might be called, are separate castings from the valve chambers, and this is the better form of construction for this type of water end; the valve chambers are placed in front of the plunger barrels and made to form a part of the supports

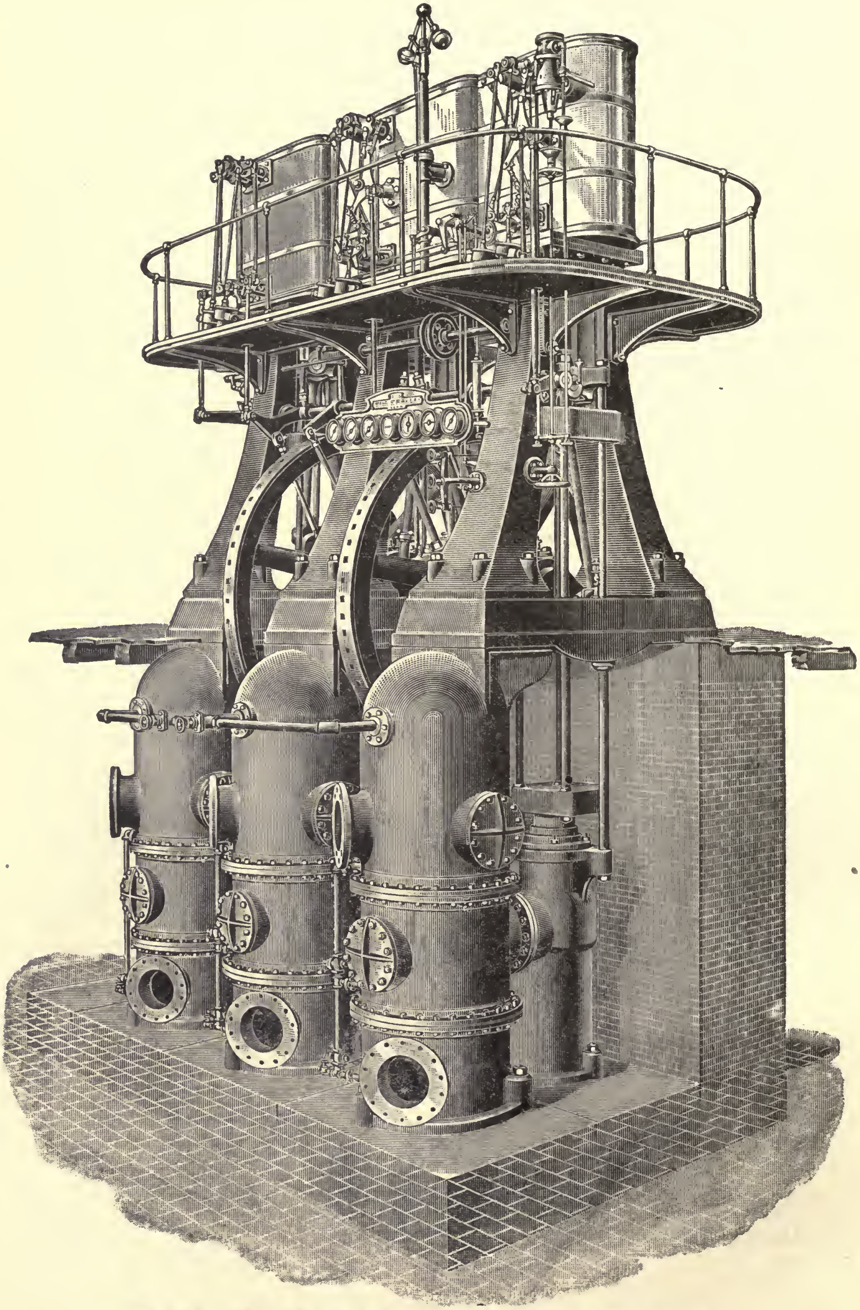


Fig 42. — Air Chambers Supporting one end of Bed Plates.

for the bedplates (see Fig. 42). The connections between the plunger barrels and the valve chambers are located so that the formation of air pockets is impossible, which is an extremely important detail not always appreciated by designers of pumping machinery, even some of those considered pretty good ones as the world goes. The course of the water should always be upward and outward without interference, after it once gets inside of the suction chamber at the bottom of the pump, and overdraughts and chances for forming air bubbles and air pockets should be avoided.

The pump work is all of the cylindrical form giving great strength and rigidity in proportion to the metal employed. The valve decks are plain flat surfaces well ribbed and supported from underneath, and the pump valves are mounted on cages as already pointed out in Fig. 36 and Fig. 37. These valves are rubber disks $3\frac{1}{2}$ inches in diameter and about one-half inch thick; they work on brass valve seats and are controlled by brass springs; the area of each set of valves, suction, and delivery, generally being about equal to the area of the cross section of the plunger. Each pump has an air chamber formed of the upper portion of its valve chamber construction, which supports the bedplates; and in some cases large equalizing pipes connect the tops of the air chambers thus formed.

The steam engine proper, or the steam power end of the machine, is practically a regular vertical triple expansion marine type of engine, with cranks, fly wheels, guides, and connecting rods; running at a low speed to accommodate the pumping of the water which experience teaches and shows cannot be profitably handled at high speed; none of the experienced and successful builders and users of pumping machinery being advocates of high piston or plunger speeds for water works engines. The air pump for the condensing apparatus is driven sometimes by an arm attached to the top end of the main plunger at the low pressure end of the engine, and sometimes by a rocker beam operated by link connections from the plunger head; the air pump driving mechanism also drives the boiler feed pump,

jacket water pump, small air compressor pump, etc., so that there are no auxiliary apparatus to be driven by steam from the boilers.

The connections between the steam pistons and the water plungers are by means of the usual piston rods, from pistons to main cross heads, sometimes two rods and sometimes one rod to each piston; and then from the main cross head there extend four heavy tie rods to the pump plunger, thus transmitting the motion of the piston directly to the plunger without the intervention of links, beams, or similar details, and making the machine direct acting in all essential effects, the mechanical efficiency of the larger machines going as high as 96 per cent in some cases. The steam piston packing is usually composed of cast iron rings joined where the ends of the rings come together by a brass "keeper," often two rings in each piston, nearly square in the section of the ring, and backed by light, thin, steel springs of just sufficient tension to hold the rings against the cylinder walls.

The steam valve gear is of the Corliss type on the high pressure and intermediate cylinders; and on the low pressure cylinder sometimes a combination of Corliss steam valves and poppet exhaust valves is used, while in some of the engines all poppet valves are used on the low pressure cylinder; and sometimes the intermediate cylinder is fitted with Corliss steam valves and poppet exhaust valves. The clearance or waste room in these cylinders is often brought as low as 1½ per cent in the high pressure, 1 per cent in the intermediate, and less than one-half per cent in the low pressure. The high pressure cut-off is controlled by the usual fly-ball governor for the top speed limit, combined with a variable hand adjustment. The intermediate and low pressure cut-offs are adjusted by hand and set where the operation of the engine indicates the best efficiency in actual service. These engines have seldom been, if ever, used on the direct or closed system of pipes, and at the worst upon rather large systems or upon systems having a stand pipe, and for the reason that they have generally been

built of the larger capacities, because it is doubtful economy to produce and use triple expansion high duty engines upon a small scale; but of course the larger capacities mean extensive distribution systems even where there are no reservoirs, and large systems mean a very considerable percentage of the pumping capacity of each engine utilized, which in turn means a pretty free delivery and ease of pumping. Therefore, beyond taking care of the top speed limit, and affording reasonable adjustment where the natural and automatic demand and supply do not properly respond to each other, there have been no practical attempts made at close pressure regulation; but of course there is no reason whatever why a pressure regulator cannot be applied to the vertical triple as well as to other forms of pumping machinery.

Since the design and introduction of the vertical triple expansion, crank, and fly wheel pumping engine in 1886, the original builders have produced about fifty engines having an aggregate capacity per 24 hours of 600,000,000 U. S. gallons, other builders having followed this design more or less, having quickly recognized its good qualities, especially its excellent fitness for use in large sizes, say from 10,000,000 to 50,000,000 gallons daily capacity. Aside from the original builders there have been produced and placed in water works about thirty of these engines by other builders in different parts of the country, and it bids fair to hold a very prominent place for a long time to come, as there are no signs at the present time of any great advances in any other direction in the line of economically pumping water for public supply.

CHAPTER XIII

VARIOUS TYPES AND CLASSES

A reasonably complete list of pumping engines which have been or now are regularly built for water works, including the four pronounced types referred to in the last chapter, would be as follows:

Non-Rotative or Worthington Direct Acting Type

Compound non-condensing, horizontal.
Compound condensing, low duty, horizontal. (Original Worthington.)
Triple non-condensing, horizontal.
Triple condensing, horizontal.
Compound condensing, high duty, horizontal.
Triple condensing, high duty, horizontal and vertical.

Rotative or Crank and Fly-Wheel Type

Cross compound condensing, high duty, horizontal and vertical.
Double compound condensing, high duty, horizontal. (Gaskill.)
Triple condensing, high duty, horizontal. (Gaskill.)
Triple condensing, high duty, vertical. (Reynolds.)
Quadruplex condensing, low duty, inclined. (Holly.)

Beginning at the top of the list, the Worthington type of water works pumping engine has been repeated to a very much greater extent than any known form. In fact since 1863, the date of its introduction, the duplex direct acting type has been more extensively duplicated for water works purposes

than any other, not even excepting the older and original direct acting type, the celebrated Cornish engine. And since the expiration of the early Worthington patents, the shops of nearly all regular steam pump builders, wherever water works machinery of moderate sizes, say up to 5,000,000 gallons daily capacity, has been manufactured, have joined in swelling the production of this type of hydraulic machinery.

The compound non-rotative, non-condensing, horizontal machine, the first on the list (see Fig. 43), is likely the simplest form in which is it possible to construct a pumping engine at all acceptable for supplying water in a methodical manner. It of course follows the regular duplex direct acting lines, although the absence of a condenser necessitates even a lower rate of expansion than the rather low ratio of the condensing machine, but it must be remembered that the water load is added to by the one atmosphere pressure of steam which must be driven out of the low pressure cylinder when no vacuum is present to help the work along. There are two high pressure cylinders and two low pressure cylinders for the steam end, and generally one large casting contains the two water plungers. It has the steam pistons, pump plungers, and duplex steam valve gear constructed practically the same in principle as in the larger and more refined classes of this type. It is not very extensively used for water works purposes, upon the score of lack of steam economy, although the machine may be bought for comparatively a low price. However, in some cases, say up to 1,000,000 gallons per day, where coal is very cheap and plentiful, this class of engine is occasionally employed.

The second on the list, the compound condensing, horizontal, non-rotative pumping engine, is fully described in Chapter IX as the original water works engine of this type.

The third on the list, the triple non-condensing, horizontal machine, is practically the same as the first mentioned at the head of the list, with a third pair of cylinders added so as to use the steam three times instead of twice. It is not much used in water works because the compound condensing does

more with its steam at almost the same or perhaps a little lower cost for plant including boilers.

The fourth on the list, the triple condensing, horizontal, known as the "low duty triple," is the machine third in the list, with its cylinder proportions slightly changed and a condenser added. This form of the non-rotative machine up to 6,000,000 gallons daily capacity, makes a very good compromise between low and high duty for small and moderate sizes of pumping engines. Cities and towns pumping enough water would be justified in purchasing a high duty compound condensing engine, but the economy of fuel is of course just as desirable in the smaller plants as in the larger ones, and it is perfectly plain that high duty may be too expensive, for its real value must be based upon the saving in fuel which it can effect, as balanced against the capital account represented by the interest on the cost of the engine, and also the cost of the maintenance of the engine in good order and repair. The time came when the low duty compound condensing machine was left too far in the background by the high duty standard machine to hold its place, and hence a step forward within the reach of small and moderate buyers became necessary. The "low duty triple" meets the requirements in a very satisfactory manner.

The idea was to produce a pumping engine so simple in design that its cost would be comparatively low, and yet contain elements which, although not by any means ranking with the cut-off high expansion machines, nevertheless would show a saving in coal that would justify a slightly greater price, when pumping a small quantity of water, than the low duty compound could be bought for. It must also equal the compound practically, in low cost of repairs.

As already pointed out, this class of the Worthington type is a six cylinder machine at the steam end, having the cylinders in tandem pairs with the low pressure cylinder at the outer end. See Fig. 44 for a general view of the machine, and Fig. 45 for a longitudinal section of the steam end. A good deal of study

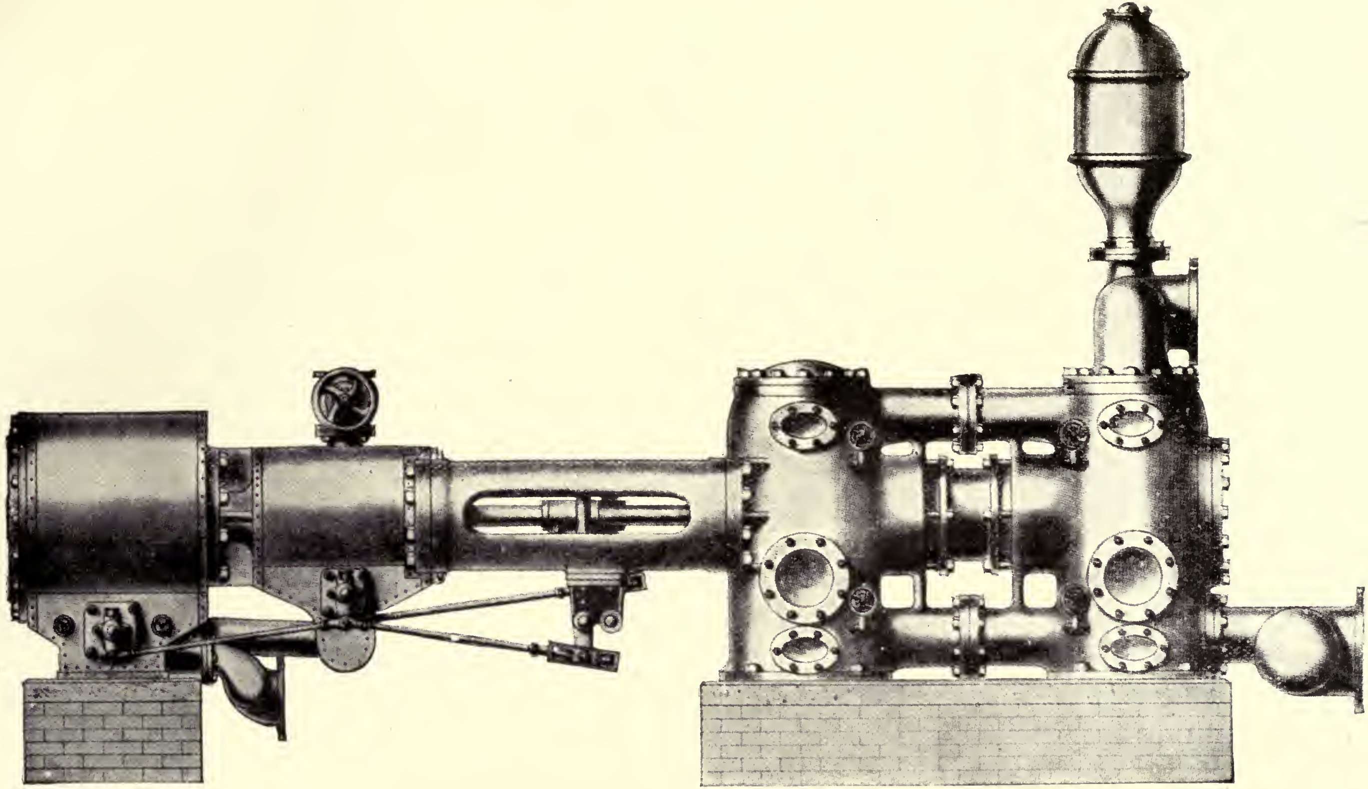


Fig. 43. — Compound Non-Condensing Worthington Engine.

was given to this engine with the factor of accessibility in view, and the successful arrangement for getting at the various pistons is apparent in Fig. 45 which is a horizontal section through the cylinders. Each piston and cylinder may be examined by the removal only of the respective cylinder head. The high pressure and intermediate pistons may be removed between the high and intermediate cylinders; while the low pressure pistons come out at the back end of the machine in the usual manner.

The steam valves are of the Corliss type and located beneath the cylinders, thus giving a very perfect drainage and practically preventing any moisture entering the cylinders at all, such moisture going directly into the exhaust ports at once; and it has been demonstrated in practice that these engines do not need the usual drip valves on the steam cylinders. Cut-off valves are fitted to the high pressure cylinders only, and the mechanism is arranged to cut off from the half to the three quarter point in the stroke. The valve gear is very simple, and in principle like the regular duplex, modified of course to suit the Corliss valve, and driving directly to the valve arms.

The drive of the main pump at each side of the engine is accomplished by attaching the plunge rod to a cross head extending across the framing, so as to connect with the single piston from the high pressure and the pair of rods from the low pressure, the intermediate piston being connected by a single piston rod back through the head between the intermediate and low pressure cylinders, the intermediate piston driving directly on to the low pressure piston which in turn transmits the intermediate power to the cross head by means of the pair of rods shown.

The efficiency of this class is very satisfactory in small and moderate machines, say from 750,000 to 6,000,000 U. S. gallons per 24 hours, when pumping against ordinary water works pressures of from 40 to 100 lbs. per square inch. The duty of the engine varies according to size and other governing conditions, from 70,000,000 to 95,000,000 ft. lbs. per 1,000 lbs. of

steam, some records going as high as 80,000,000 ft. lbs. with ordinary steam just as it comes from the every-day boiler plant without allowances for entrainment. An engine of this type and class of 2,000,000 U. S. gallons capacity per 24 hours, pumping against a pressure of about 80 lbs., has an official record on test duty of 82,000,000 ft. lbs. per 100 lbs. of coal fired, without any deductions of any kind; and its record gives a yearly duty of 72,000,000 ft. lbs. per 100 lbs. of coal for the station, including banking and heating, although it is very difficult to see what banking and heating have to do with the duty of a pumping engine.

The fifth on the list, the compound condensing high duty, horizontal direct acting pumping engine, is the engine mentioned second fitted with what is known as a high duty attachment. A clear idea of this engine is given in Chapter IX under the heading of the original Worthington pumping engine.

The sixth on the list, the triple condensing, high duty horizontal and vertical, is a still further development of the non-rotative type of pumping machinery, and represents the largest and most economical pumping engines of this type. It is produced in capacities ranging as high as 40,000,000 U. S. gallons per 24 hours, and has a record of 175,000,000 ft. lbs. of work per 1,000 lbs. of dry steam.

The high duty horizontal triple condensing, direct acting or non-rotative type is shown in Fig. 46, and the vertical triple engine is illustrated in Fig. 47, the latter indicating a class of this type employed in capacities of from 20,000,000 to 40,000,000 U. S. gallons per 24 hours.

The seventh on the list is the cross compound condensing, high duty crank and fly wheel pumping engine, which is built in both the horizontal and the vertical form. By cross compound is meant that the steam is transmitted across the engine on its way from the main steam pipes to the condenser, expanding and doing work on its way. There is a high pressure steam cylinder at one side of the machine, which takes its supply of steam directly from the boiler and exhausts into a

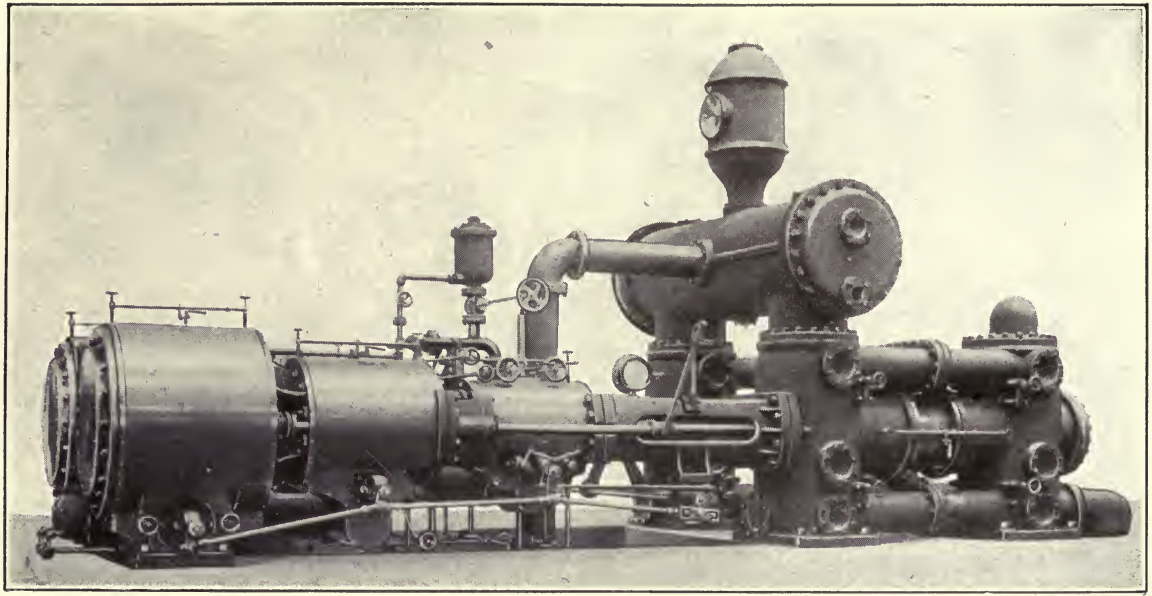


Fig. 44. — Low Duty Worthington Triple Expansion Pumping Engine.

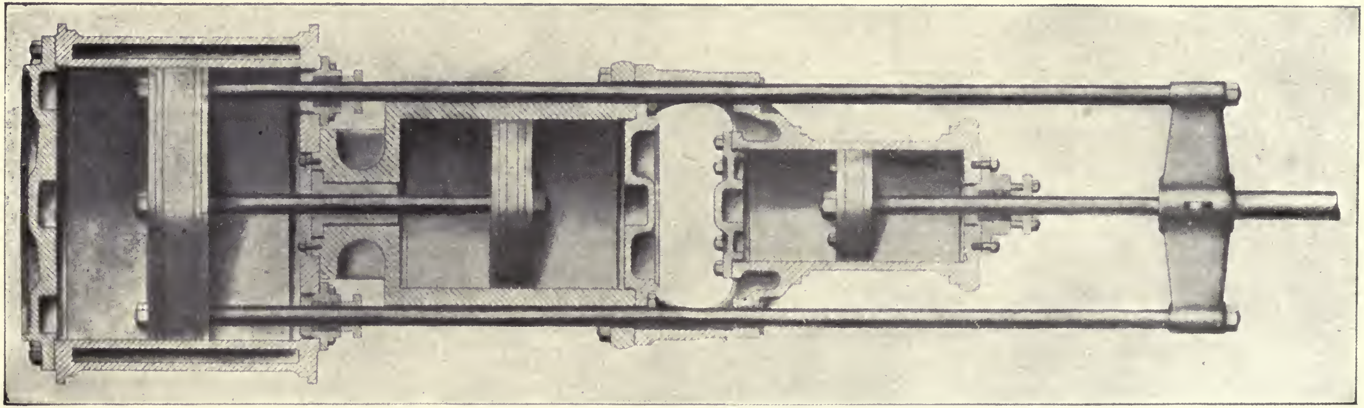


Fig. 45. — Section Through Worthington Triple Cylinders.

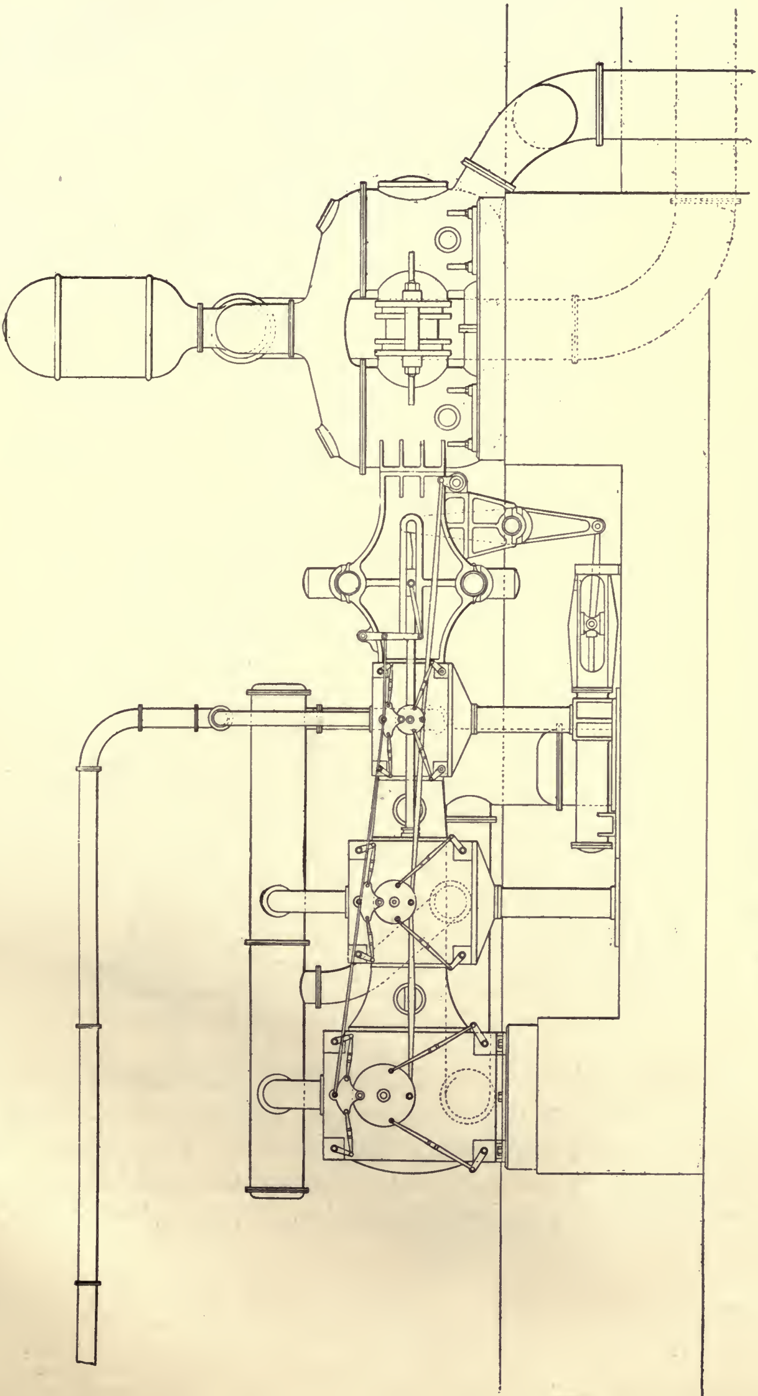


Fig 46 — Worthington Horizontal High Duty Triple.

receiver; and a low pressure steam cylinder at the opposite side of the machine, taking its steam from the receiver and exhausting into the condenser. This type of engine would be impossible for practicable operation in pumping, without the use of

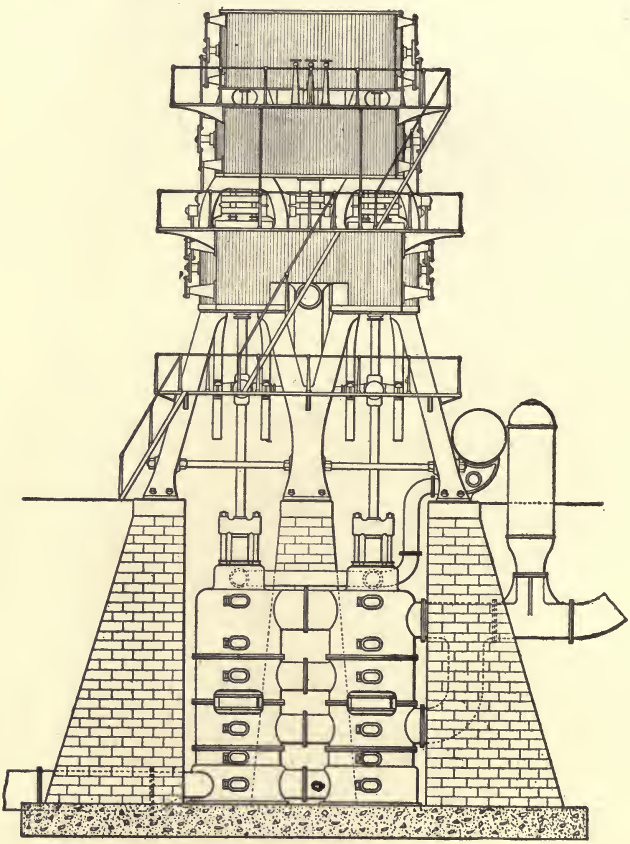


Fig. 47. — Worthington Vertical High Duty Triple.

the crank and fly wheel for connection and regulation; it is a very good type of pumping engine where it suits the conditions, but there are many cases in water works where the unbalanced operation of the two sides of the machine make it undesirable. Also where the steady and full load of reservoir

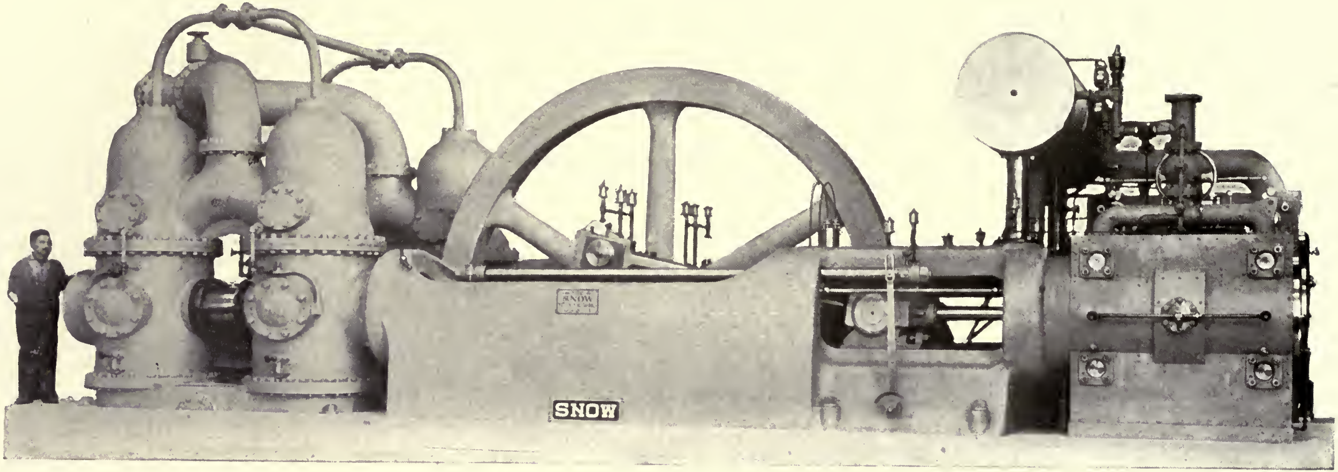


Fig. 48. — Snow Cross Compound Horizontal Pumping Engine.

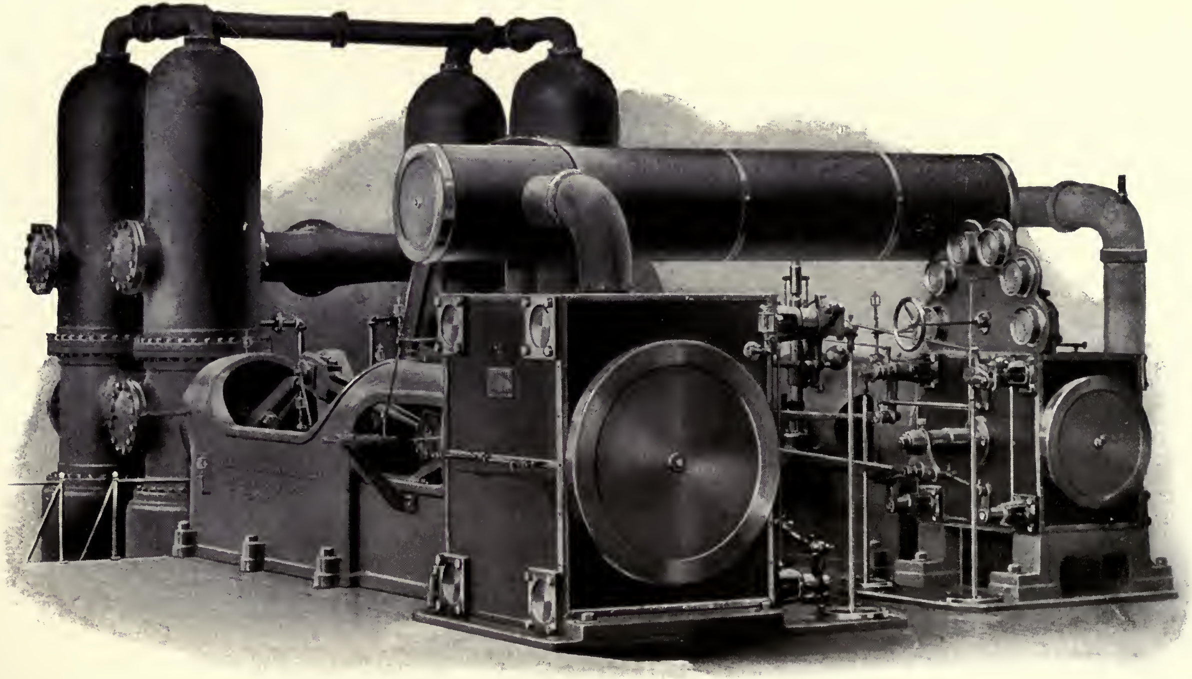


Fig. 49. — Allis-Chalmers Cross Compound Horizontal Pumping Engine.

service or an equivalent load is encountered, the use of the cross compound engine does not give the buyer that full benefit of steam economy that might be just as well obtained by the use of a triple expansion machine, especially in cases of con-

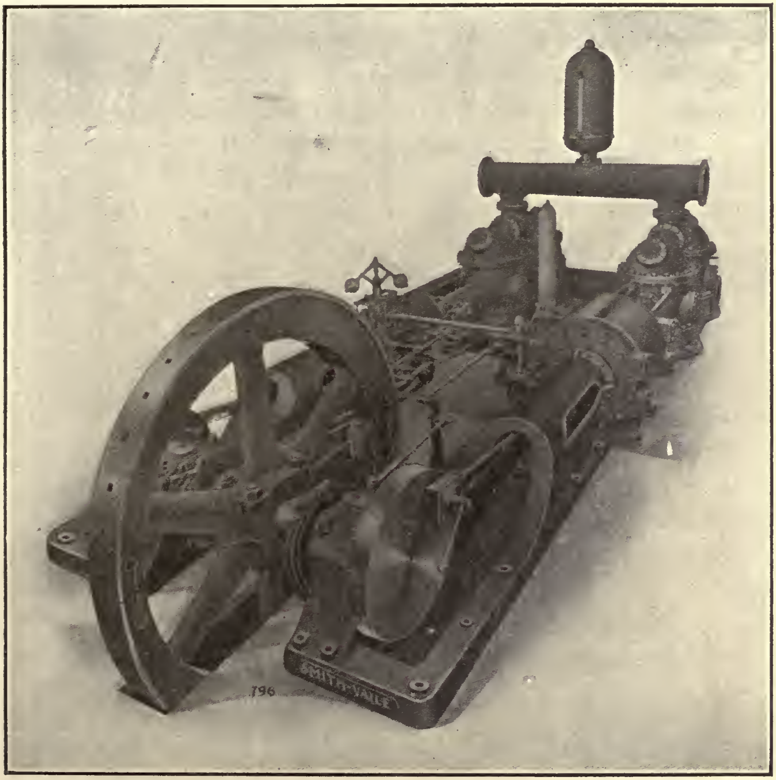


Fig. 50.—Platt Iron Works Company Horizontal Cross Compound.

siderable capacity, say from 10,000,000 to 20,000,000 gallons and upward per day.

The horizontal cross compound pumping engine originated in the attachment to a regular cross compound Corliss mill engine, of two water pumps, one behind each steam cylinder; the plungers being driven by an extension of the steam piston

rods through the back steam cylinder heads, and then prolonging these rods into plunger rods entering the water cylinders. It is no doubt the simplest form of the horizontal crank and fly wheel pumping engine, but on account of very considerable space required has not been extensively used in water works plants, especially where new engines are placed in old buildings.

The Platt Iron Works Company of Dayton, Ohio, has followed out this idea in a substantial and consistent manner, and has also taken an important step apparently in advance of most of the others so far as concerns reduction in the cylinder waste room or clearance, by placing the steam valves across the cylinder heads in this type of pumping engine. The water plungers are very easy of access at the free end of the water cylinders; the steam pistons are reasonably accessible especially when it is considered that the modern steam piston requires comparatively little attention; and the main pillow blocks, cross heads, and connecting rods are as easy to get at as in a Corliss mill engine. This company seems to be making some special and apparently successful efforts in adapting the Corliss type of steam end to various types of pumping engines, including direct acting or non-rotative machinery, as may be seen in Fig. 56, which shows a design of compound condensing engine, and Fig. 57, which shows a four cylinder, cross-triple engine.

The Snow Steam Pump Works of Buffalo, N. Y., reduced the cross compound horizontal pumping engine practically to a regular although not frequently repeated type, by placing the crank shaft between the steam and water ends of the machine, the steam pistons and water plungers being rigidly connected together by means of tie rods which pass above and below the crank shaft. The Allis-Chalmers Company of Milwaukee, Wis., also produce a cross-compound engine of this type, and the originality of design will as a matter of course be disputed between these two firms. The machine has a great advantage by being accessible as to pistons and plungers, at the outer ends of the steam and water cylinders, and reasonably so with reference to the main pillow blocks and connecting rods. There

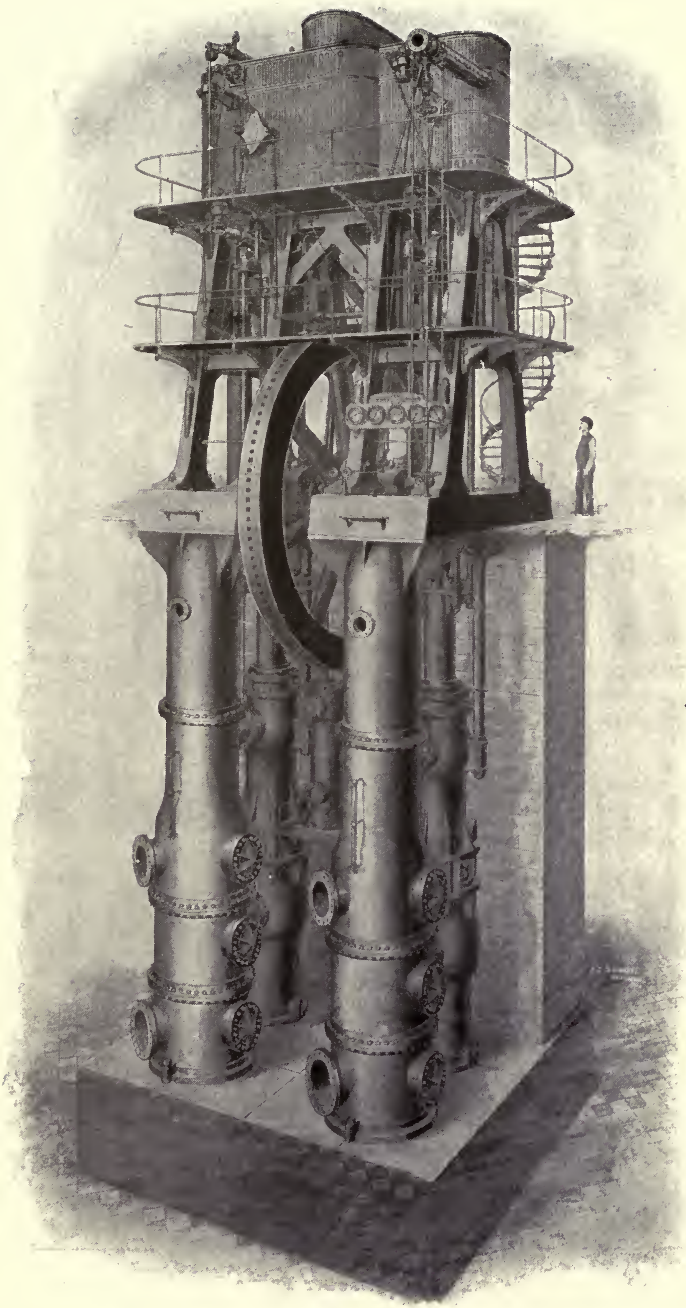


Fig. 51. — Allis-Chalmers Vertical "A" Frame Compound.

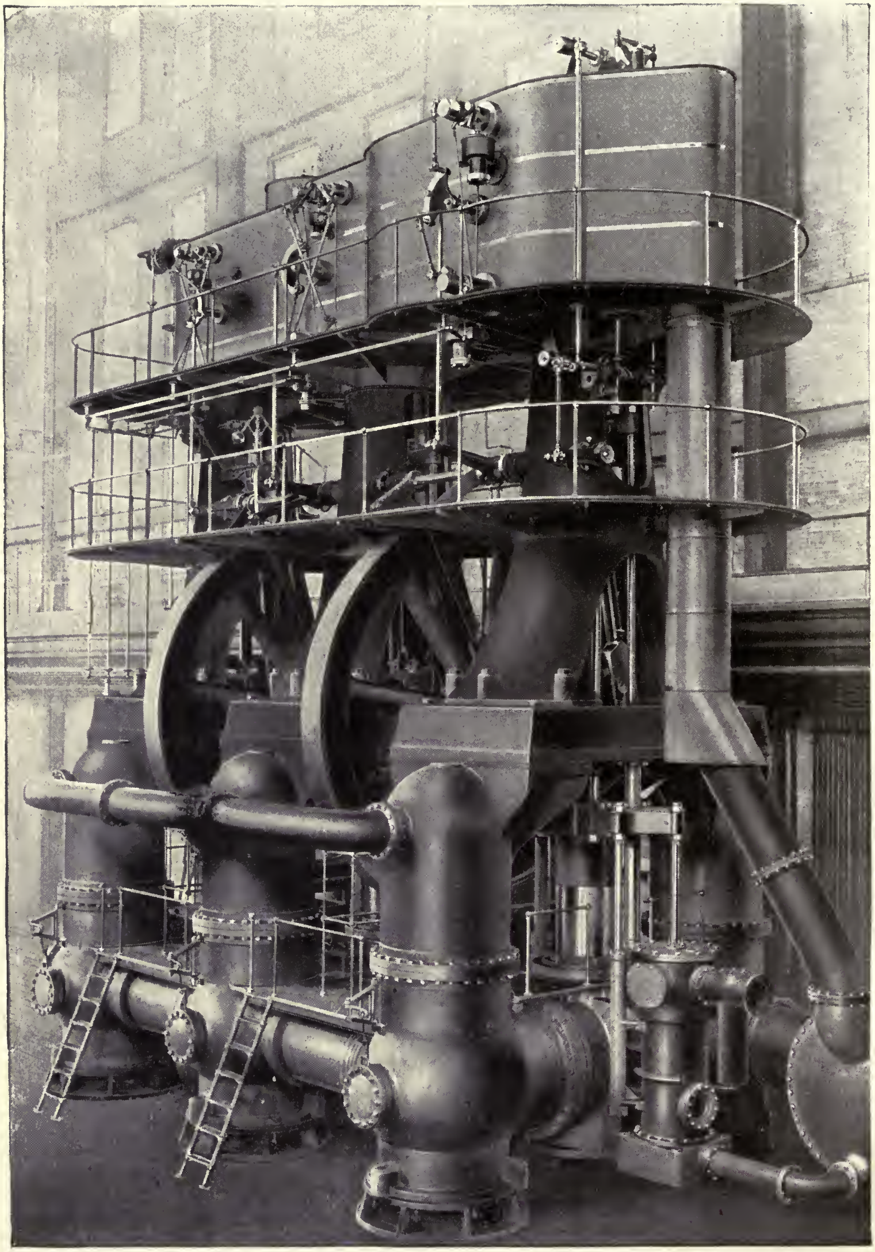


Fig. 52.—Allis-Chalmers Vertical Tower Frame Compound and Triple.

have not been very many of these horizontal cross compound machines built, most likely for the reason that there are not so very many places where they will fit in economically, all things considered. It seems as though the cross compound came too late into the field; the gap between the low duty triple and the high duty triple is comparatively narrow, and is well taken care of by the regular and thoroughly introduced types of high duty compound machinery such as the Worthington and the Gaskill-Holly engines.

Fig. 48, Fig. 49, Fig. 50, Fig. 51, and Fig. 52, show different makes of horizontal cross compound pumping engines, and also two different makes of vertical cross compound machines, all designed and built by leading engineers and manufacturers. As to economical results, there is no reason in the expanding of steam in two cylinders provided with appropriate cut-off mechanism, why this form of machine within proper limits will not accomplish as high a record as any other type, of a similar number of expansions, and similar circumstances. In fact the record does show that the results are about the same with compound machinery under equal ratios of expansion and mechanical efficiency, similar steam pressures and work accomplished.

There has been much talk about high ratios of cylinders in compound engines, competing with the triple engine, in steam economy; but the secret of efficiency in high ratio compound cylinders is the unusually large low pressure cylinder incidentally afforded, and the consequent facility of adjustment of the engine to its load. But the excessive range of temperature that appears when too much steam expansion is attempted in any one cylinder will always prevent two cylinders reaching the economy of three cylinders. And aside from this, the many advantages in using three single acting plungers in a pumping engine would be enough to dispose of all arguments in favor of cross compound engines, where the cost and capacity are favorable to the triple.

The eighth on the list, the double compound, condensing, high duty horizontal, crank and fly wheel pumping engine, is

fully described in Chapter XI as the Gaskill engine, and enjoys the distinction of being the first high duty crank- and fly-wheel pumping engine, probably the first high duty pumping engine of any type, regularly built as a standard or commercial machine.

The ninth on the list has been of limited use in water works plants. It is the Gaskill compound engine with an additional pair of steam cylinders, these latter cylinders forming the high pressure cylinders of a six-cylinder triple machine of the horizontal, crank and fly-wheel type. (See Fig. 53, which gives a very clear idea of this machine.) Its limited employment in pumping stations is no doubt due to the idea that when the step to triple expansion is taken, the fact presents itself that instead of stopping half way at a compromise point, it is better to adopt the complete triple machine, the vertical triple, three-cylinder engine, and thus simplify the machine and increase the efficiency at the same time, even though at slightly greater cost.

The tenth on the list is the Reynolds vertical triple expansion engine, now rapidly taking the lead as the ultimate development of modern high duty pumping machinery, already having just about reached the extreme possibilities in expanding steam under practicable working pressures. This engine is described in Chapter XII in fairly good detail.

The eleventh on the list, although now obsolete and out of the market, the Holly quadruplex pumping engine, especially designed for closed systems of pipes, is well worthy of mention. It held an important place in the public water supply field twenty-five years ago, and was well thought of as a standard machine in its day. A description of this type of machine is given in Chapter X as one of the distinct types of pumping engines.

There are in addition to the foregoing, several offshoots from the Worthington type of high duty non-rotative machinery, two only of which seem to be of sufficient prominence to call for special mention — the d'Auria and the Groshon engines; and only one of these, the d'Auria, has been repeated sufficiently

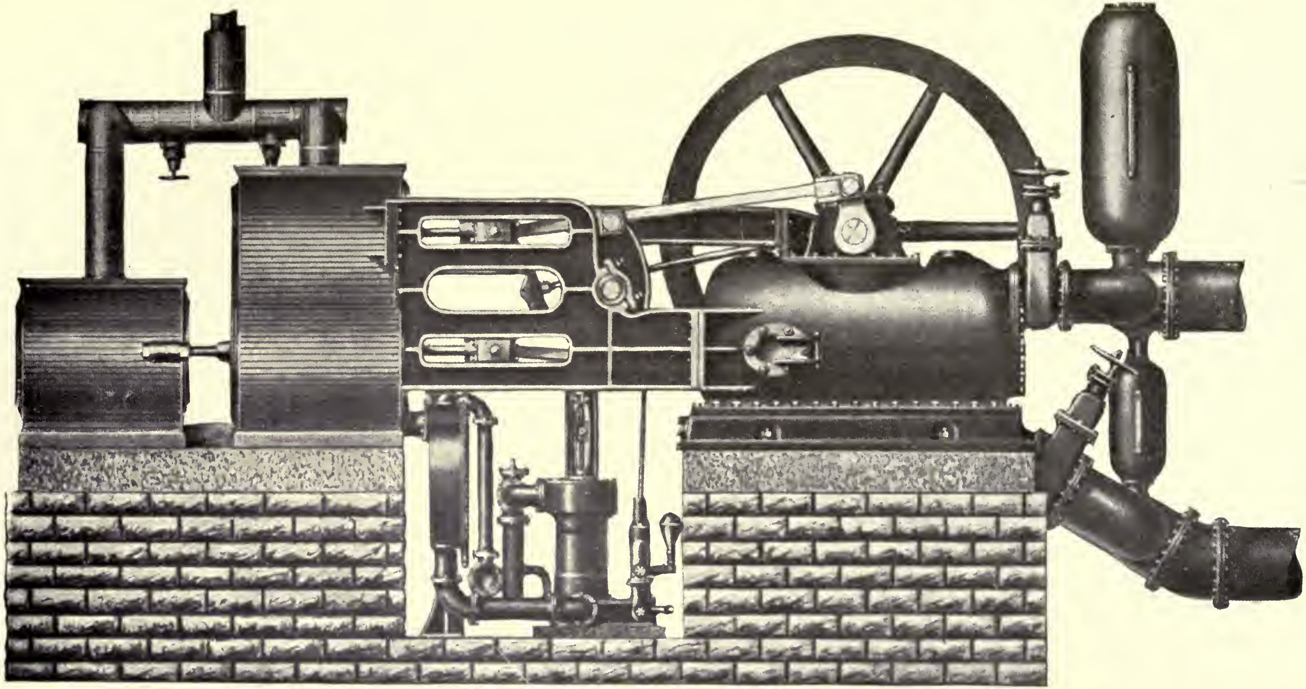


Fig 53. — Gaskill Horizontal Triple Pumping Engine

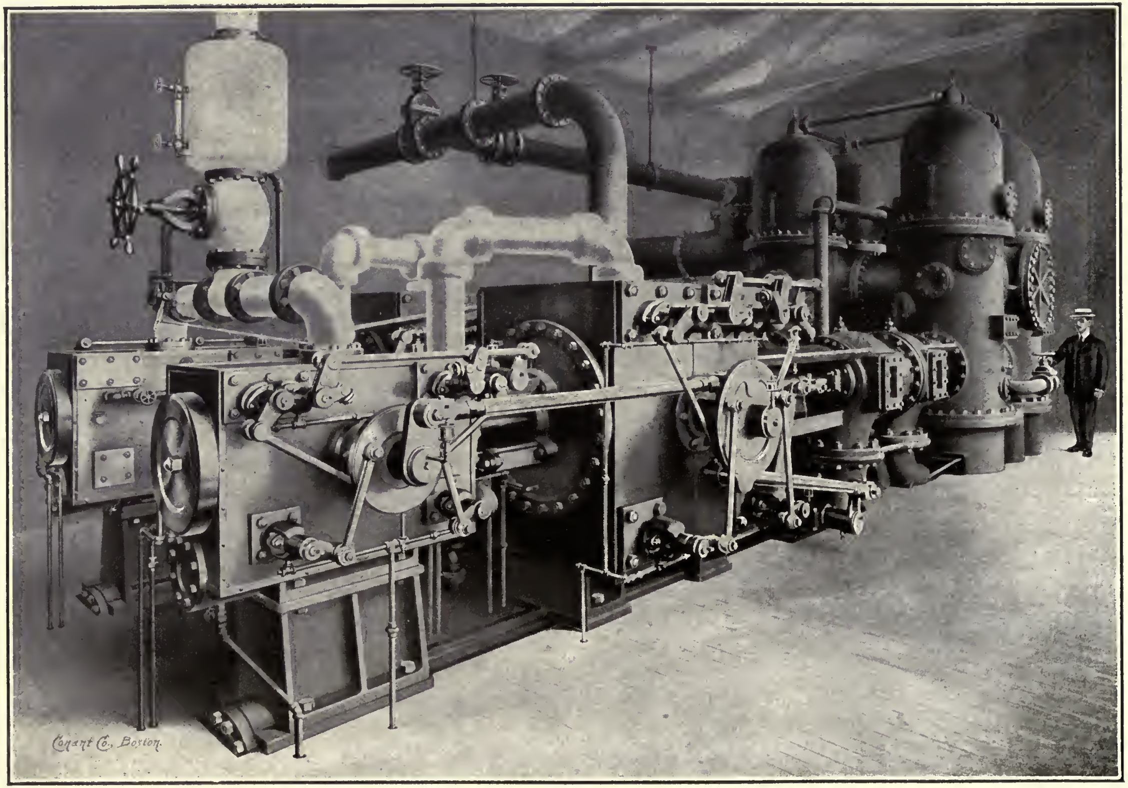


Fig. 54. — General View of d'Auria Pumping Engine.

to entitle it to a place as a regular pumping engine for public water supply. Both of these engines would come under the fifth and sixth in the list, that is, of the non-rotative type and belonging to the high duty compound and triple classes, as they can both be produced either as compounds or triple expansion machines, precisely upon the lines of construction followed by the Worthington machinery, with differences in the compensating devices and valve gear.

The d'Auria high duty pumping engine is practically a Worthington machine fitted with a balancing or compensating device of the hydraulic variety, and consisting of a liquid col-

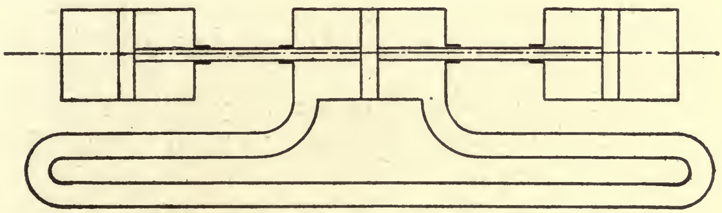


Fig. 55. — Hydraulic Loop of the d'Auria Pumping Engine.

umn contained within an inclosed loop or pipe, connected with a chamber which contains an auxiliary or compensating plunger. This plunger is attached to the same rod that carries the main pump plunger and the steam pistons of the pumping engine. (See Fig. 55 for the hydraulic loop, and Fig. 54 for a general view of this class of the non-rotative type.) When the engine is working, the compensator or auxiliary plunger forces the liquid in the pipe loop into an alternating or pendulum-like motion. During the early part of the stroke, before the cut-off, the surplus energy due to the initial steam is absorbed mostly by the hydraulic column in the loop, to be given out again during the latter portion of the stroke when the expanding steam has become too weak to drive the load. Of course the object is, as in the case of the fly wheel of the crank engine, or the compensating plungers of the Worthington

engine, to equalize the inequalities of steam expansion throughout the strokes of the engine.

This class of the non-rotative type is being built for water works pumping stations in sizes from 1,500,000 to 10,000,000 U. S. gallons per 24 hours, and in general design is exceptionably good from a mechanical standpoint. The castings are of unusually good design; and so far as can be seen, this engine if properly placed upon a commercial basis and pushed with the energy with which some of the others have been handled, would no doubt get its share of business; for no matter how good the inventor or designer may think his engine is, or good it really is, the public practically knows nothing about it, and it requires a good deal of persistence, patience and hard work to demonstrate any certain lot of facts in competition with some other facts opposing.

The Groshon high duty non-rotative pumping engine is a pumping engine of the Worthington type fitted with Groshon's valve gear and compensating device, the valve gear being an adaptation of the Corliss releasing gear with dash pots; and the compensating device is a combination of auxiliary water cylinders, levers, and connections, so arranged as to make use of the water pressure in the main discharging pipe, to equalize the inequalities of the steam expansion. The machine seems to be practicable, but very few have been built and put into service; although, as in the case of the d'Auria engine, commercial energy properly and liberally applied would no doubt result in the use of this engine to a corresponding extent. Fig. 58 shows a general view of the Groshon pumping engine above the floor line, and Fig. 59 shows a side elevation exhibiting the working parts of the valve gear and compensating device.

Aside from the regular types and classes of pumping engines for water works which have gradually taken their places from time to time, there have been quite a number of attempts at special designs which have been seldom or never repeated; and such engines have no doubt been fondly looked upon by their designers at the time of production, as the perfection of

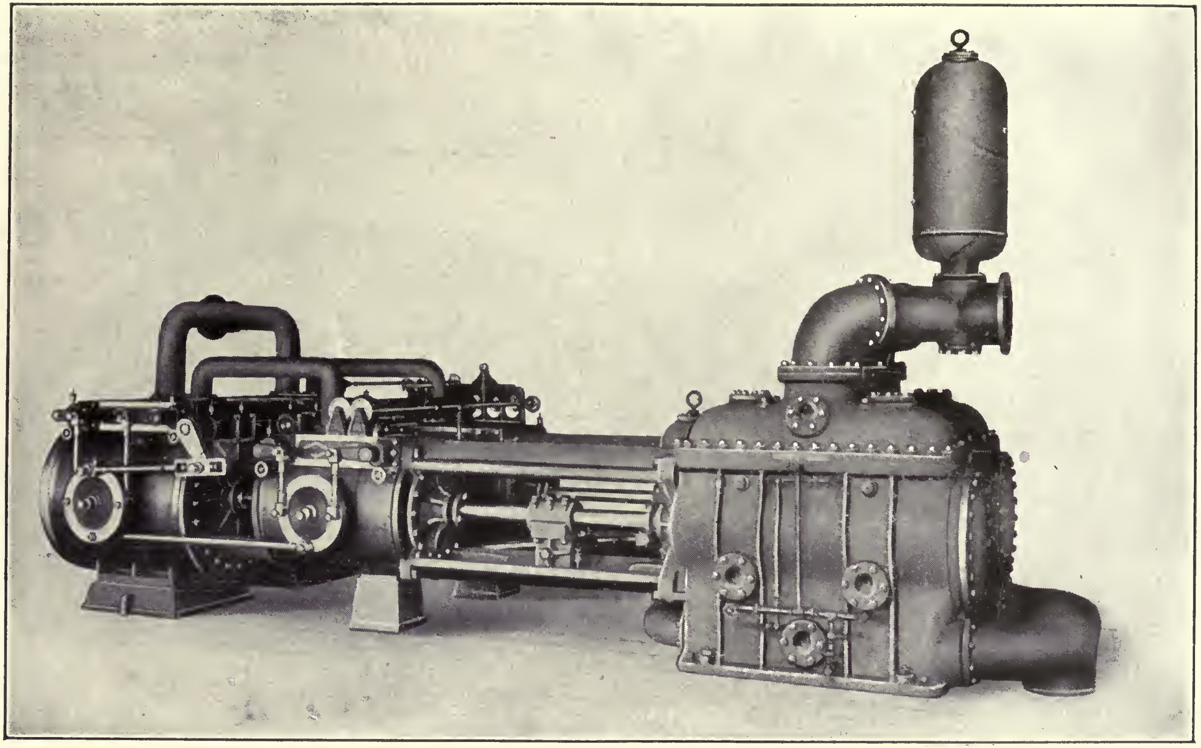


Fig. 56.—Platt Iron Works Co. Compound Direct Acting Pumping Engine.

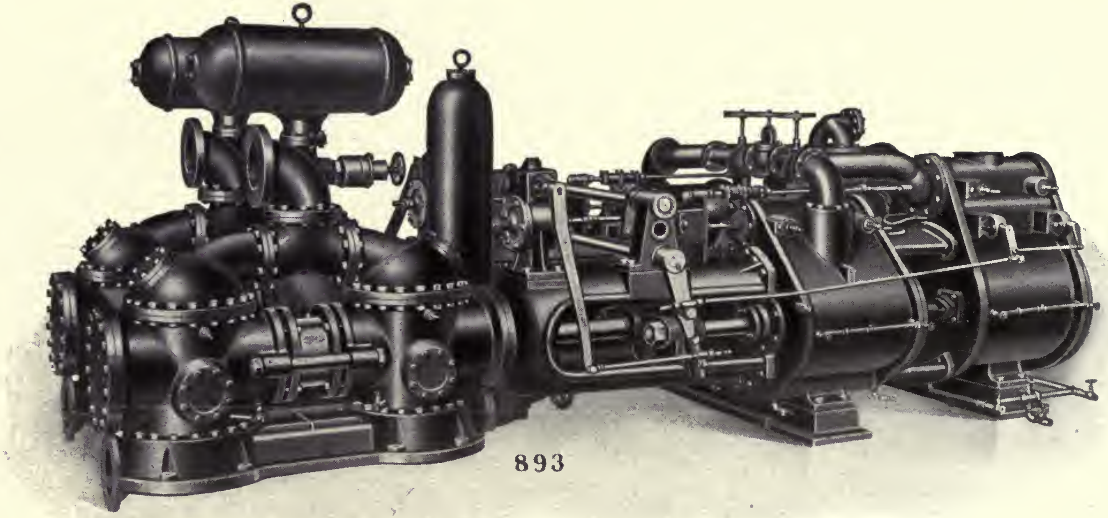


Fig. 57. — Platt Iron Works Co. Cross Triple Direct Acting Pumping Engine.

accomplishment, to be copied and repeated by engineers for all time to come. Of course all new machines which represent radical departures from well-trodden paths are of special design, and the attempts to make them successful and admirable are extremely earnest and sincere. But it has been observed that the most valuable and enduring advances have been made along the lines of evolution more or less slowly; the few brilliant and marked examples now and then taking their places as a sort of punctuation in the general progress.

And so it may be noted that during the years of development of what may be called commercial pumping machinery, although it was not considered as such in the early days, for water works, from 1885 to 1892 occasional designs of a special nature with the aim of much better steam economy mostly in view came out from time to time; and no doubt these special engines spurred the makers of the regular products to greater efforts. Long and heated arguments were indulged in by the different advocates of various ideas, all tending to fix the lines of average, where capital and coal accounts could meet upon common ground, and where the buyer could make the best investment all things considered.

The more notable special pumping engines giving duties above 100,000,000 ft. lbs. per 1,000 lbs. of steam, were the Corliss Pawtucket cross compound, horizontal engine of 1878; the Leavitt Lawrence compound beam engine of 1879; the Reynolds Milwaukee compound beam engine of 1881. These engines, however, although clearly demonstrating the possibilities of refinement along the old lines of steam jacketing, high steam pressures, steam expansion, condensers, and air pumps, were too costly to justify their general adoption in water works service, even though they did show their ability in the line of much greater steam economy.

The Corliss Pawtucket engine is shown in Fig. 60, the Leavitt Lawrence engine in Fig. 61, and the Reynolds Milwaukee engine in Fig. 62.

Among the older and interesting pumping engines which

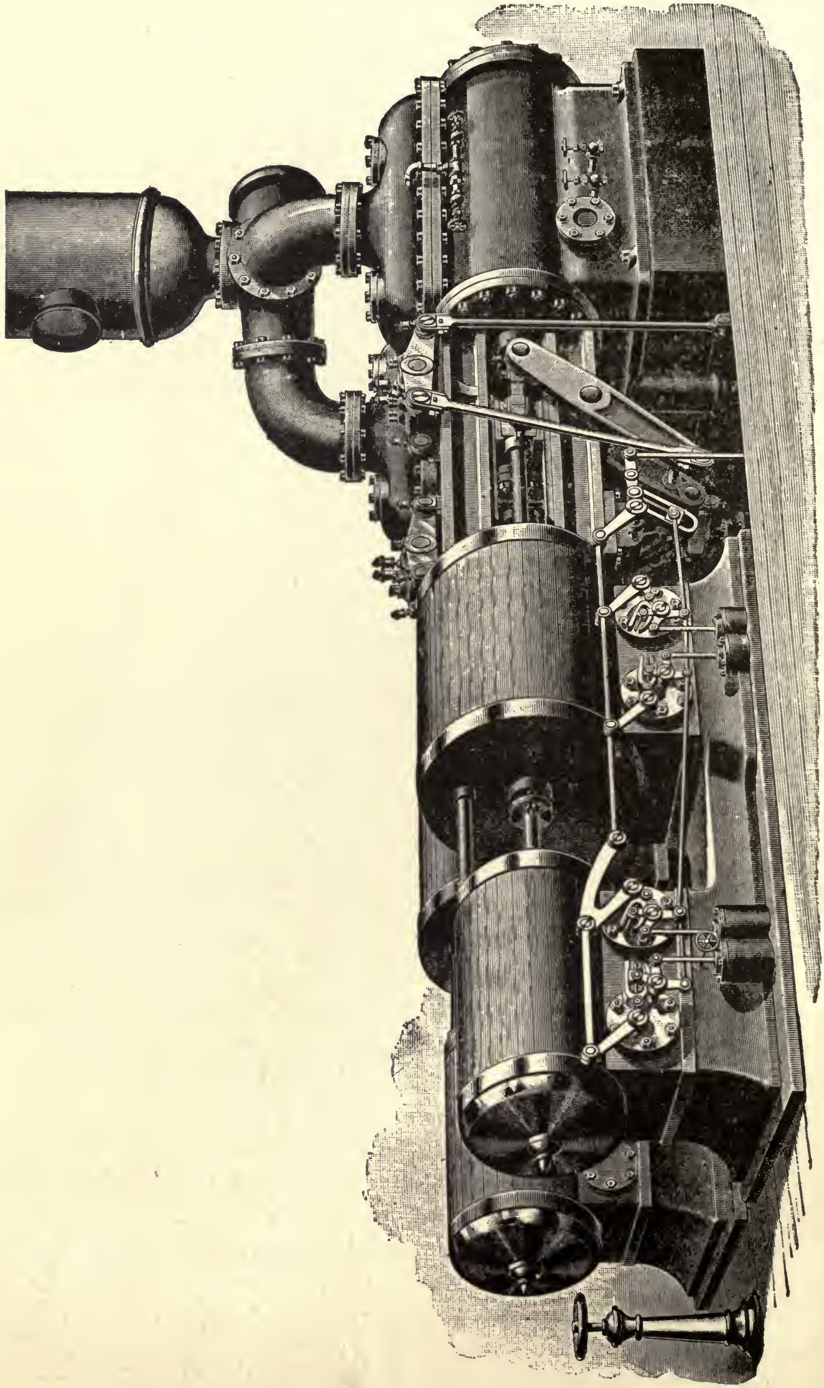


Fig. 58.—General View of Groshon Pumping Engine

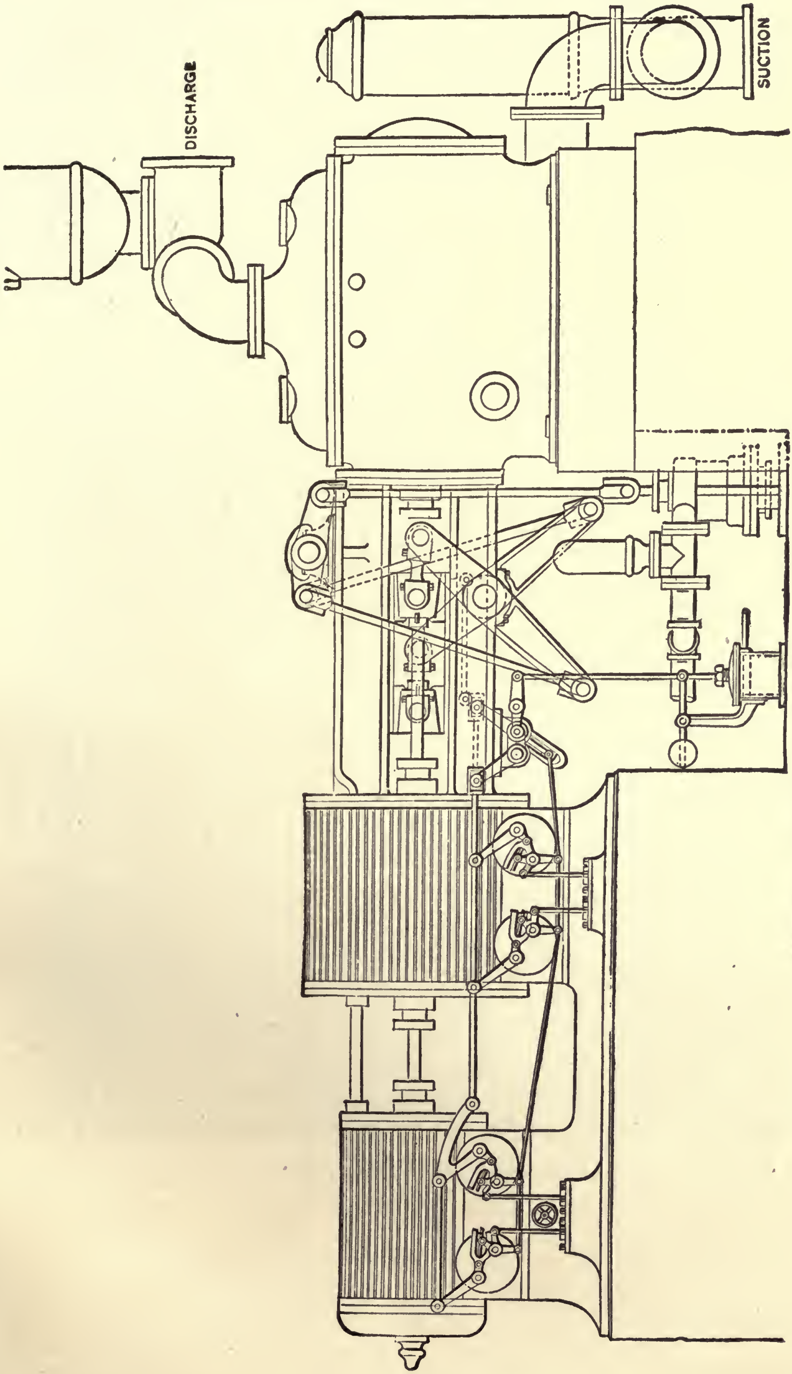


Fig. 59. — Side Elevation of Groshon Pumping Engine.

have never been repeated on account of cost and other objections, but in which it was sought to excel the performance of the Cornish engine of 70 years ago, and previous to the advent of the so-called "high duty" engines already referred to, there might be mentioned the Shields engine, a Bull Cornish

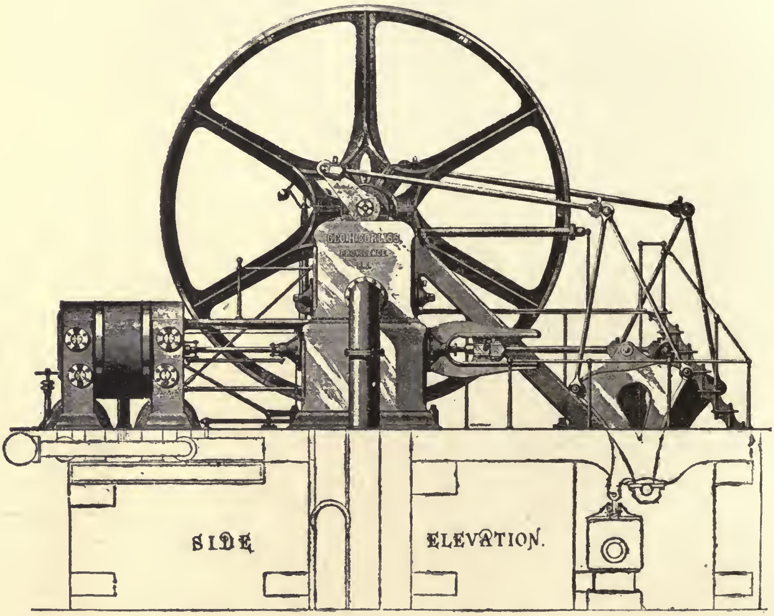
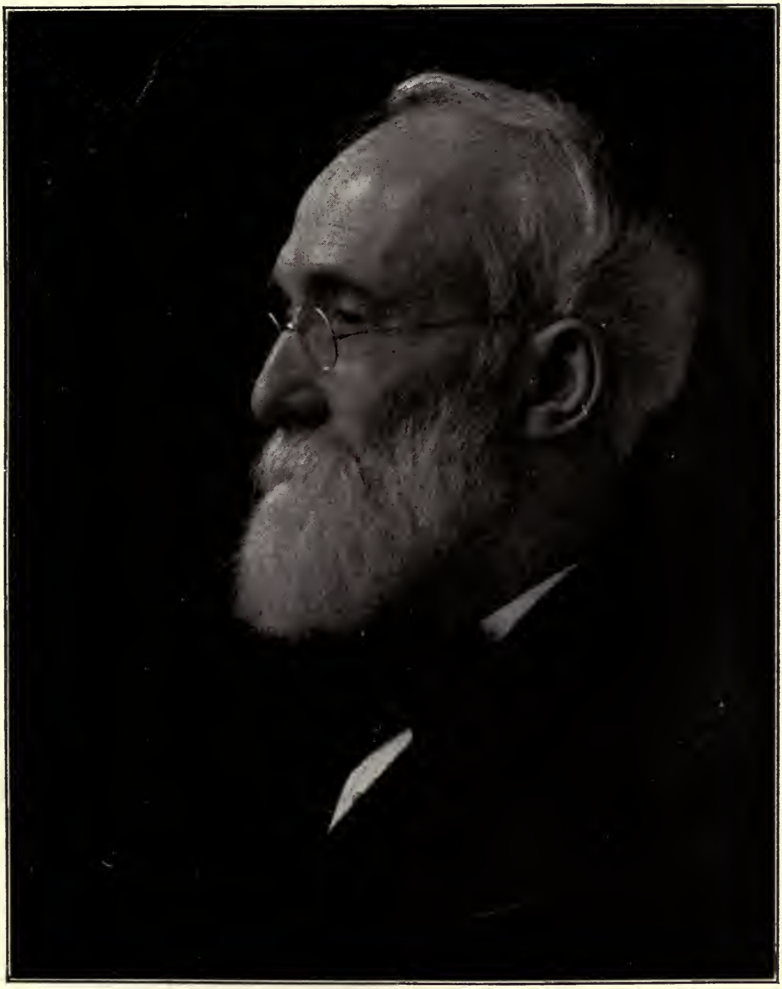


Fig. 60. — Corliss Pawtucket Pumping Engine, by George H. Corliss.

as nearly as it resembled anything, at Cincinnati, Ohio; the full Cornish engine of Shedd at Providence, R. I.; the Lowry engine at Pittsburg, Pa. And there were a few others hardly of enough importance to mention here and which perhaps it will be better to permit to sleep on in unknown resting places, although they no doubt pointed a useful even if painful lesson at the time of their design and production.



ERASMUS D. LEAVITT.

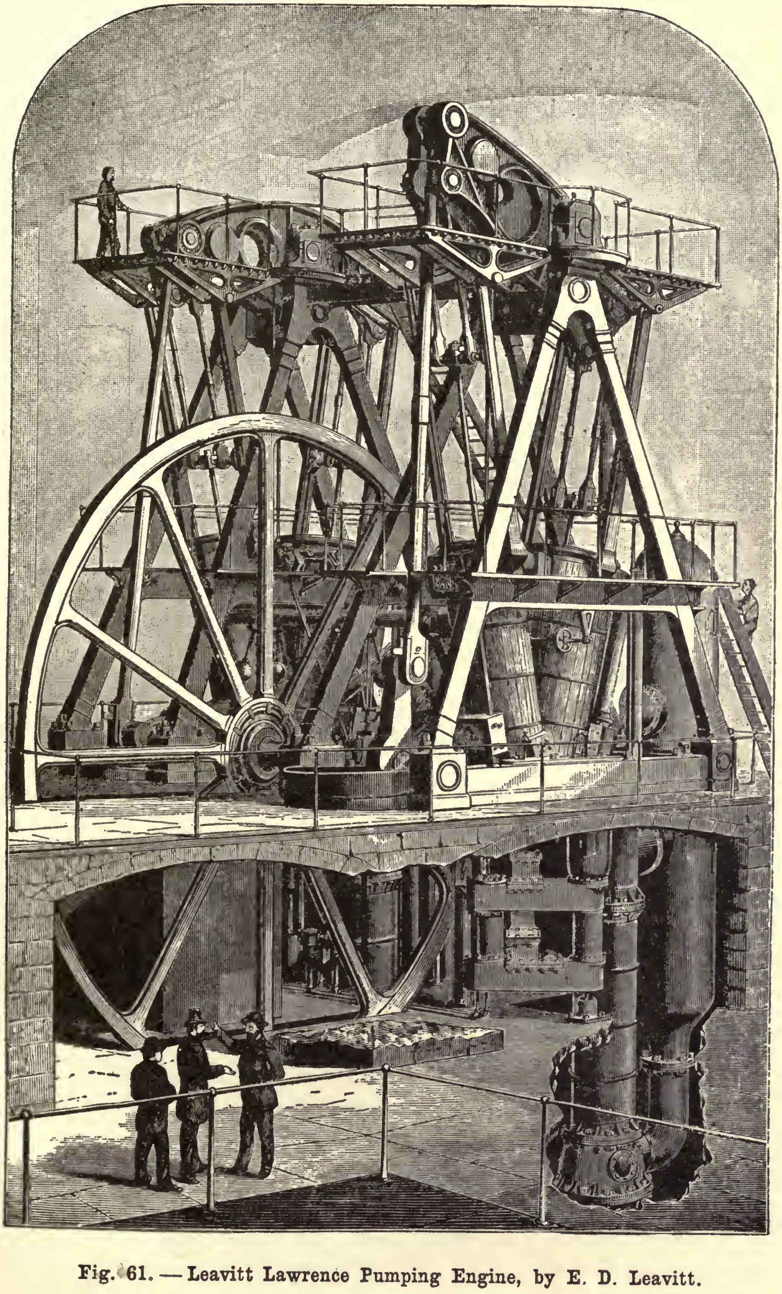


Fig. 61. — Leavitt Lawrence Pumping Engine, by E. D. Leavitt.

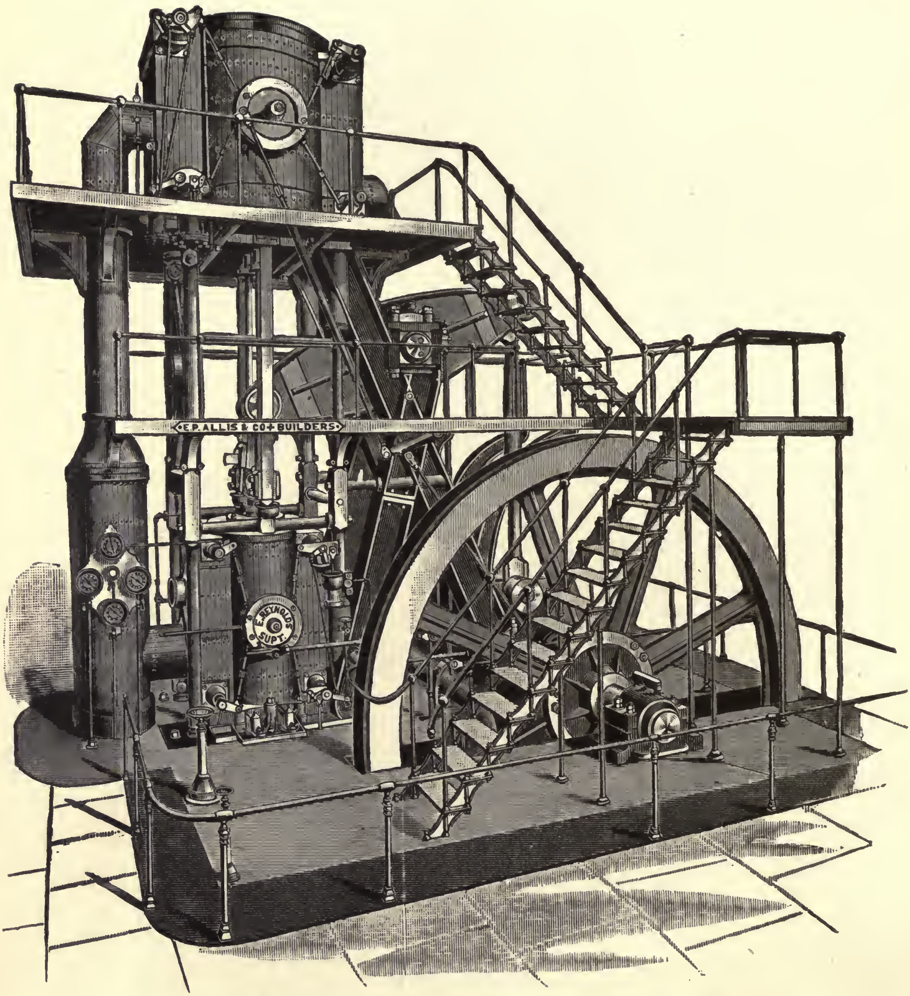


Fig. 62.—Reynolds Milwaukee Pumping Engine, by Edwin Reynolds.

CHAPTER XIV

PUMPING ENGINES ADAPTED TO CONDITIONS

THE proper adaptation of a pumping engine to the conditions to be met in service, is the key to the highest practical efficiency. Reservoir work, or the work of pumping only into a storage or distributing reservoir, and with the force main having no connection whatever with any of the distributing systems, is no doubt the easiest and most economical task that pumping machinery is called upon to perform. But even at that, there are as simple as the proposition seems at first sight, some important items to be carefully looked out for. In the matter of steam economy alone, there are needs of reasonable considerations in connection with other details. There are advocates of high piston speed, of high steam pressure, of high rate of revolution, and other singled out and isolated factors. But the general combination wherein the machine best meets the conditions is what will yield the best results, and not the exploiting of any particular seemingly important factor by itself. And as an example of this it may be noted that the pumping engine in this country, if not in the world, which a year ago, April, 1906, held the high duty record per 1,000 of steam, has the following set of conditions to work under:

Capacity per 24 hours, 15,000,000 U. S. gallons.
Piston speed, 197 feet per minute.
Rotative speed, 19.7 revolutions per minute.
Water load against pumps, 126 lbs. pressure, 290 ft. head.
Steam pressure per gauge, 126 lbs.
Indicated power, 802 horse power.
Mechanical efficiency, 96 per cent.
Steam per hour per indicated horse power, 10.68 lbs.
Duty per 1,000 lbs. of steam, 179,450,250 ft. lbs.

With reference to high piston speed, the best record known to the writer as to pumping engines is 607 feet per minute, where the duty per 1,000 lbs. of steam was 157,843,000 ft. lbs., showing that high piston speed alone will not answer.

With reference to high steam pressure, the record seems to be 200 lbs. per gauge and a duty of 149,500,000 ft. lbs., showing that high steam pressure in the absence of other ruling conditions or proper fitness, falls short of the best performance.

Regarding thermal efficiency, or the actual economy of heat employed with reference to absolute temperatures, even the greatest thermal efficiency does not in the presence of adverse conditions in some other directions, enable the engine with a very high thermal efficiency to equal the engine working under a better general fitness of things, as will be seen by the following:

Thermal Efficiency.	Duty per 1,000 Lbs. of Steam. *Foot Pounds.*
22.80	149,500,000
21.63	178,497,000
21.00	179,454,250
20.85	173,620,000
20.78	176,419,600

What has been considered for some time the world's record for general all around efficiency for a pumping engine is as follows:

Capacity per 24 hours, 30,000,000 U. S. gallons.
Steam pressure per gauge, 185 lbs.
Piston speed, 195 feet per minute.
Duty per 1,000 lbs. of steam, 178,497,000 ft. lbs.
Duty per million heats units, 163,925,300 ft. lbs.
Steam per indicated horse power per hour, 10.335 lbs.
Thermal efficiency, 21.63 per cent.
Mechanical efficiency, 96 per cent.
Rotative speed, 17.73 revolutions per minute.

The complaint has often been made that pumping engines do not give so high a duty in regular service as during the official test with experts. In general terms this might be admitted and because the experts know how to adjust the engine better and how to operate it more closely to conditions found to exist than the running engineer usually found in pumping stations; and this difference in operation in favor of the expert is because the expert, by his more extensive knowledge of the subject, is a more valuable and a higher paid man than the regular attendant. But it would not pay for all the actual and practical difference in economy, to employ a scientific and professional expert to operate a pumping station, although for purposes of contract comparisons the very best work of the engineer is sought to be brought out by the contractor, and this is just what the expert does, leaving it to the regular running engineers to approximate as nearly as they can under every-day conditions and the many cares of the situation, the pace set at the official test. But aside from the fine adjustments which the expert is able to make in the engine so as to completely adapt the machine to its work at the time of the test, while the engine cannot be expected to keep up to the fine adjustment in the hands of its regular attendants, any material falling off in duty can be traced to changed conditions.

For reservoir work exclusively, consisting of pumping to a reservoir through an independent main, and in the absence of service pipes for supplying consumers, the fullest liberty can be allowed in selecting the type of pumping engine to be used; and any reasonable type or class in the use of which the economy of steam and coal would be satisfactory might be installed. But when the service of supplying consumers comes into the question, and where the water from the pumping station is sent through street distribution pipes, the types of machinery must be restricted to those which will give reasonable smoothness in operation and a great deal of freedom from pulsation. The system not infrequently met with of having the distributing mains supplying the consumers situated

between the pumping station and the reservoir, the general consumption satisfying its demands and the surplus going on into the reservoir located somewhere beyond the main distribution, furnishes a case in point; and a two plunger pumping engine with 180 degree crank pins will require excessive air chambers to enable it to give any decent approach to satisfactory service; in fact any type or class of rough working engine will under such conditions cause a great deal of trouble and complaint. Going a step further, we have the closed system, often known in some parts of the country as the "Holly System," where a closed circuit of pipes is used without storage or free delivery, and in fact without any delivery aside from the consumers' supply pipes and the inevitable leakages; in such a system of course the pumping machinery should have the smoothest possible delivery, and be capable of the closest practical automatic regulation. Steam economy takes a third or fourth position under the closed circuit, and certainty of pumpage supply, steadiness of pressure, and promptness in responding to fire alarms are the most important factors in the system. Such systems are mostly used in the smaller cities found in the flat regions of the middle west; those of fairly good size, say from 25,000 to 100,000 population usually have stand pipes in connection with what would otherwise come under the head of closed systems. Where the closed system of pipes becomes of considerable extent and especially where stand pipes are attached to such systems, the pumping engines can be proportioned so that they will divide the work up into units of quantity, and then one or two engines might be operated practically under reservoir conditions so far as pumping nearly or quite up to capacity is concerned; but the ever present liability of fire, which will call for an immediate and great increase in the pumpage, it being remembered that there is absolutely no storage aside from what is in the source of supply and the pump wells, requires that the running engines must be kept well within their capacity so as to be available for quick increase in their delivery; reserves being kept in cons-

stant readiness to supplement the machinery working on the domestic supply if the demands for fire service become urgent.

It might be appropriate to this branch of the subject to mention types and classes of pumping engines best adapted to the various conditions of service; they would be as follows:

Pumping Engines for Reservoir Service Only.

Horizontal engines with two double acting plungers.
Vertical engines with two single acting plungers.
Vertical engines with two differential plungers.
Vertical engines with one bucket and plunger.
Vertical engines with four single acting plungers.
Vertical engines with three single acting plungers.
Direct acting vertical engines with two double acting plungers.

Pumping Engines for Reservoir and Distributing Service.

Horizontal engines with two double acting plungers.
Direct acting vertical engines with two double acting plungers.
Vertical engines with three single acting plungers.
Vertical engines with four single acting plungers.

Pumping Engines for a Closed System of Distributing Pipes.

Horizontal engines with two double acting plungers.
Vertical engines with four single acting plungers.
Vertical engines with three single acting plungers.
Direct acting vertical engines with two double acting plungers.

In buying and installing a pumping engine, there seems to be a certain procedure which has come to be adopted during the gradual development of the business, and although shorter and bolder courses may at times be followed, upon the whole it will no doubt be found that the natural evolution in the

matter is a pretty safe guide to follow. It has seemed to some at times as though the mere statement of the requirements to be met was all sufficient for the attraction of competition, and that the would-be buyer could be furnished with all the necessary information and details relating to what he was to get for his money, by the competitors themselves. This seems beautifully simple, but there are certain attributes in human nature which completely defeat most of such efforts. Each bidder wants the work; and as the object and aim of competition is to get the most and best for the money, the result is that the party with the lowest bid tries to convince the buyer that his machine is upon equal terms with the others, and he has the lowest figures; while a higher or even the highest bidder tries to convince the buyer that he has very much the best proposition, although at a higher figure or the highest figure. The buyer's best course may be upon a middle ground somewhere between the highest and lowest; but he does not know if unfamiliar with the minutiae of the subject; and when he sees the figures varying 100 per cent from the lowest to the highest for what he supposes ought to mean the same thing, he is entirely at sea, and begins looking about him for some sort of help out of his dilemma.

The wide open call for proposals for a pumping engine will result in this state of affairs sometimes, because certain vital details are not specified in the call for bids, and different bidders will offer machinery to be run at all sorts of speeds, the main object seeming to be to make the lowest price; and, although the lowest price accompanied by proper conditions, adequate dimensions, etc., ought to be the one chosen, it behooves the buyer to be certain that he is safe in his selection. Of course this brings up the question of professional or expert advice in making a proper selection, and the logic of this is that it will be better to have competent specifications prepared in advance, than to endeavor to select a proper proposal from the "grab bag" collection liable to develop from the open call.

The matter of specifications would seem at first glance to be

very simple, and perhaps it is, to those builders who understand, and have the courage to offer what they really know ought to be furnished; but the ever present grind known by the name of competition, coupled with the strongly grounded idea that a contract should go to the lowest bidder, will assert itself and interfere with the fairly rational treatment of the subject. Aside from this there is a tendency amounting to a determination at times, to stipulate that the builder of pumping engines must provide all subfoundations, foundations proper, do excavating, cut into and replace masonry, floors, walls, and whatever may be changed in the course of the installation of the machinery; thereby inflicting upon the maker of machinery a lot of work entirely and completely outside of his legitimate business and occupation. The writer holds, and is encouraged by experience in the belief, that the best course for a buyer to pursue with reference to his own interest is to exempt the engine builder from all work and responsibility outside of the machinery itself, even to the painting of it, leaving him only that which he is prepared to handle, and so leave his mind free from the haunting shadows of matters foreign to his business and which he cannot meet without a certain element of uncertainty as to cost, and the reflection of which most surely comes back to the buyer in the shape of increased price by reason of percentages added by the machinery maker to cover possible contingencies of which he cannot accurately inform himself beforehand.

Let us map out a course which would be followed, indeed has been followed, with very satisfactory results, whether the purchaser be a private corporation or individual, or a city hedged about by legal requirements in the making of a contract. First it should be stated in some sort of an announcement that the proposals will be received at a certain hour, date, and place; and putting all upon an equitable footing with sealed proposals. General data should be stated for the information of the intending bidders, arranged in a convenient form so that those at a distance need not go amiss in

preparing a proposal, or put under the actual necessity of sending some one to investigate. Cases of course differ; sometimes the new machinery is to go into an old building, and sometimes a new building is to be provided for its accommodation, so it may not be practicable to lay down a hard and fast rule; but the general data from an actual case may give some idea of the requirements:

Static water pressure at the level of engine room floor.
Static water pressure from floor to level of water in well.
Allowance added for friction in force main.
Allowance added for friction in suction main.
Total working load on the plungers.
Steam pressure at the engine throttle.
Available clear height above the engine room floor.
Engine room floor to basement floor, vertically.
Available distance across engine room.
Wall to wall of engine room, inside across.
Available on floor lengthwise of engine room.
Available in basement lengthwise of engine room.
Distance from building wall to pump well.
Air chamber required at inboard end of suction pipe.
Air compressor required for force main air chamber.

It also seems to be desirable to state at least closely approximate, the required length, or the stopping place on the contractor's part, of the suction, delivery, and steampipes, and in these and other items remove to the utmost extent any and all uncertainties, so that the builder of pumping engines may be able to figure upon some exact basis. In fact, a clear and comprehensive statement of the work which the buyer wants done will help greatly in the matter, and save a great deal more money than it will cost to make such a statement. A general outline of the work to be done, modified of course by conditions in different cases, would call for the making of the design; furnishing general and detailed drawings or blueprints; and erecting in the pumping station upon foundations to be

furnished and built by the buyer, in accordance with the detailed blueprints, templates and anchor bolts, furnished by the engine builder; a pumping engine of the desired type and class, together with appurtenances, connections, piping and fixtures within the engine room, complete and ready for continuous service.

A call for detail drawings or working plans of a pumping engine by a buyer is not looked upon favorably by some bidders, as it is considered to be rather an exposure of trade secrets; but in some cases of public work there is a legal requirement to the effect that all details of a contract must be made public and submitted, at least so far as the Board or other public body officially making the purchase is concerned. At any rate, even if the detail drawings should be suppressed, a clever draughtsman could produce upon paper a very close imitation of any machine in operation, which might be open to public observation, and which as a rule water works engines are. And in addition to this opportunity for copying, the technical journals publish, from time to time, notable specimens of all kinds of engines and machinery nearly enough in detail to afford a pretty good guide as to capacity, strength, and detail of construction.

Some of the first things needed to be known by the builders are the conditions of service under which the engine is to operate. This can be conveniently made known by stating the number of gallons required to be pumped in twenty-four hours, giving the total water load upon plungers, including friction of suction and delivery mains, static head, point and manner of the delivery of the water, and making this statement in a positive manner, thus releasing the engine builder completely from all responsibility for the details making up the total load, but requiring of him an engine capable of delivering the given quantity of water against the aggregate working pressure under the stated conditions.

In the matter of design of a pumping engine, under specifications drawn closely enough to indicate the type desired

clearly and unmistakably, the actual design may be well left to the builder, the stipulation being made that the engine if to go into a building already in existence, must conform reasonably to its proposed surroundings, so that proper and convenient space about its various parts may be assured. In the matter of adapting this or that general type of machinery to any particular work or service, perhaps the buyer might profit by independent expert advice, so as to guard against the well meaning but misguided zeal of makers possessed with strong desires of selling machinery which might not possibly represent the very best investment for the buyer, but which might incidentally give its maker and advocate some trifling advantages over competitors in making a contract.

The length of stroke is a very important governing detail in fixing the cost of a pumping engine, as so many other details hinge upon this one; and there is no good reason for not stating the stroke of the pistons and plungers, and their speed as well either or both in feet of travel and revolutions per minute giving the allowance of excess desired in the plunger capacity. The buyer might just as well place all competitors upon an equal footing at once, leaving very little to argument or uncertainty, and he will find that much the easiest and more economical way of dealing with the matter. Subterfuges relating to percentages of plunger diameter to length of stroke are sometimes indulged in for the purpose of stipulating dimensions, but by far the better way is to come right out with the desired proportions of the water end of the machine, conforming of course to good practice, but leaving the steam factor largely to the builder on account of the duty guarantee generally required. The writer has observed cases wherein it would have been much better to appoint the experts before buying the machine than to wait until after completion for the regulation test of the machinery; and for the reason that even where the designing is left to the builder, certain stipulations covering principles of construction and proportion if properly carried out will insure the results asked for by the

buyer, and all the experts would need to decide would be whether or not these stipulations had actually been met; a test much more readily and decisively accomplished than some of the ends aimed at during the test after construction.

For example, we know that a pumping engine at a given speed with certain diameter of plungers and length of stroke, will, when properly made, displace just so much water, the construction and workmanship being the guide as to capability. We also know from records and experience that certain proportions in the construction of steam cylinders and appurtenances will perform safely certain economical efficiencies; the entrainment of water in the steam, or the leakage of a force main between the engines and the reservoir, will not have any bearing upon these facts as points of design and construction in the engine. If the buyer is suffering from bad boilers and force mains, the engine builder cannot help him out by modifying the machinery. Give the engine builder dry saturated or superheated steam for his engine or the equivalent allowances therefor, and then take the water away from his pump under the stipulated load, and that is as far as he can fairly be held responsible for results.

The various parts of the machinery should be of plain and substantial design with appropriate shapes and forms adapted to the special purpose; the principal aims being for ample strength, great reliability, good mechanical effects, etc. Where the design is made by the builder, and the work is in competent hands, there is not very much to say, but there is no harm for the buyer or his representative to know that the bedplates and framing will be designed so as to prevent loss of alignment, or undue strains, or changes of load distribution, from changes of temperature or other causes; castings so designed as to avoid objectionable changes of section with reference particularly to shrinkage strains; reinforcement of the body of the casting next to the flanges, proper fillets, rounded corners, reëntering angles, and all such details which may just as well be had at the same price as something less desirable; the de-

sirable machinery coming as much or more from a thorough knowledge of design and construction, than from an advance in price asked by the builder.

The general construction and arrangement of the pumping engine will of course depend upon whether it is horizontal or vertical, as either one of these distinctive types will follow lines peculiar to itself; also depending upon whether the type will be of the crank and fly wheel or the direct acting variety, that is to say, rotative or non-rotative. In drawing up specifications for pumping engines upon the part of the buyer, it is not good policy to go too far in the direction of actual design or even dimensions; but rather to set forth the conditions and requirements to a pretty exact degree, and by so doing the competition will be kept within certain restrictions, a good close and real competition on figures will be obtained, and a great variety of bids on machines most of which would not be wanted, kept from complicating the efforts to secure the type or class of machine really wanted and best adapted to the work to be done in the current service of the water works plant. The writer can recall a recent case, where in ten or twelve bids from different concerns representing the best manufacturers of pumping machinery in the country, with business headquarters situated hundreds of miles apart, the dimensions given by the bidders for the machinery proposed to be furnished were alike in a large majority of the offers, and the few which differed from the majority differed but slightly. The bids reduced to equal terms did not vary more than five per cent. In the specifications under which all of the proposals were made, not a dimension excepting the length of stroke of the engine was mentioned, but the conditions of service were pinned down so closely that practically all of the bidders arrived at the same conclusion regarding the machines. A case of the opposite character is also recalled, where the writer was called in to untangle a snarl of bids; the highest being more than 100 per cent above the lowest. This condition was the result of a wide open specification of very

indefinite meaning, and wherein very few particulars were given. The bids were all rejected, and a new specification resulted in a good, clean competition with the figures about two thirds of the highest bids received under the first call, and with the different bids pretty close together.

The question as to what type of pumping engine is required needs careful consideration; whether horizontal, vertical, triple expansion, compound or double expansion, fly-wheel, direct acting, and all other matters and facts likely to have an important bearing upon the subject, should be taken into account in making a selection. In fact, the type or class of engine can be pretty thoroughly sifted down, and then the competition called for upon lines suited to the particular case in hand, and so avoid the complications and uncertainties resulting usually from the wide open specification. It is best to set forth very plainly the requirements to be met, the guarantees expected, and the terms of payment proposed; stating the various conditions under which the water is to be pumped, the duty obtained, water pressure, steam pressure, piston speed, length of duty, test, etc., etc. Also stating penalty, bonus, or damages, and the ultimate action of the buyer to be expected in case the engine fails to meet the contract, or falls short a certain amount of the contract requirements.

CHAPTER XV

INSTALLATION OF PUMPING ENGINES

The type and class of the pumping engine having been decided upon as the best for the service under consideration, of course the first question is whether the new machinery is to go into an old building, or whether it is to be a part of an entirely new plant, buildings and all. And this thought naturally enough leads directly up to the further question of foundations. And in the consideration of foundations for the support and anchorage of pumping machinery, especially for the larger sizes, it is very difficult to establish any fixed and exact rules. Many attempts have been made and much time lost in endeavoring to establish some sort of formula for reaching satisfactory results; but there are so many changing circumstances, variations in conditions, and incidental things to be taken into the account, that it is difficult to see how the question can have any theoretical side at all.

The only principle of material value is the one which involves the loading of the foundation, expressed in pounds pressure per square foot upon its bed, so as to keep within a safe working limit. And in the matter of load upon the foundation bed, it is extremely necessary to make the important distinction between a live load, such as a working engine, and a dead load, such as the walls of a building. It goes without saying, that a heavy, strong, and suitable foundation is absolutely necessary for the best results, and this is especially so in the case of pumping engines. There are all sorts of rules and ideas concerning the spread, weight, length, breadth, and depth of foundation, but the situation and conditions, as in all other matters pertaining to the accomplishment of any definite purpose, must

largely govern the efforts in this direction. A pumping engine comes under the head of "live loads" upon a foundation as opposed to anything which does not of itself embody a living and moving force; and a good general rule with reference to the foundation of a pumping engine is to allow 800 lbs. to the square foot upon the bottom or upon the bed of the foundation, inclusive of the weight of the foundation itself, when upon good earth or soil, and this will give enough area to the bottom for any good soil. With rock for a bed, a less area of bottom will be sufficient; in fact, the least area consistent with sufficient space to support the machinery at the top of the foundation will answer all purposes with a rock bed; and it is only when the work must rest upon a bed of soil, clay, gravel, or other earthy materials in the absence of rock bottom, or in the case of such extreme depth down to the rock, that the excavation and subsequent filling with foundation materials is very costly and troublesome, that the bearing pressure is to be brought down to what may appear to some as the low figure of 800 lbs. to the square foot of area of bed.

Foundations may sometimes be unnecessarily expensive on account of cut stone caps, anchor plate stones, and other costly things, but it cannot be too solid. The idea of a foundation for a living, working engine, is to make it as nearly as practicable, represent a fixed part of the entire mass of the earth by its weight, spread, and bearing; and when this is accomplished to the extent of preventing all settlements, tremor, or disarrangement of any kind, then the stability and operation of the machinery will be assured, so far as its installation and location are concerned.

Foundations for the smaller sizes and capacities of pumping machinery are generally made ample; apparently because the cost is not very great anyway, or is comparatively small and simple in form. But as the magnitude of the plant increases, the proportions of the smaller unit carried out in a larger unit, seems to make the dimensions and cost loom up to an extent which startles the designer. Of course the fact

is partly, perhaps principally, that with small machinery, say a 2,000,000 gallon horizontal pumping engine, just a monolith or solid mass of masonry of the required length and breadth, and perhaps 3 or at the most 4 feet deep, will make a proper foundation, and in such cases the weight per square foot of bed is very much less than 800 lbs. Such a small simple thing as a brick or concrete pier 10 feet long, 5 feet broad, and 4 feet deep is easy and inexpensive to build, but when it comes to a vertical, triple expansion pumping engine of the highest type and of considerable size, as, for example, 15,000,000 U. S. gallons capacity, the foundation is calculated very closely; many times too closely, to the sorrow of the buyer at some later day.

It can be stated with the certainty and authority of a self evident proposition that money invested in a proper foundation is money well invested; and that money kept out of a foundation to the extent of risking the success of the machinery, expresses a saving very badly misplaced. In fact if the question must be settled between apparent extravagance and misplaced economy, then the extravagant side of the case ought to be favored in the interests of safety, low repair accounts, and economy of operation. Each case ought to be considered and decided by itself, as experience shows that no general rule can be followed on account of the great variety of underground and underwater conditions, unknown and unseen until actually dug down to and exposed. Foundation making is not quite so purely an art as stone quarrying, but there is not so very much difference between them as at first might be thought.

When a new pumping engine is to be put into a building already in existence, a very great care is necessary to avoid doing damage to the building, its foundations, and its walls; it not infrequently being necessary to underpin the building walls before the engine foundations can be commenced. The writer in his own experience lately, had such a task in hand, and the first thing done was to excavate a pit about 5 feet in

length in line with the wall; and this pit was carried down to the bottom of the building foundation which was found to rest upon coarse sand and gravel. It being necessary to go still deeper to accommodate the new engine, which was a vertical triple expansion machine of 15,000,000 U. S. gallons daily capacity, this pit, 5 feet along the wall, was continued on down some 12 feet below the bottom of the building foundation until rock and large immovable boulders were reached. The pit was then carried beneath the building foundation for its entire thickness, and a bed of strong Portland concrete was laid 3 feet in depth, and upon this concrete base a pier of brick laid in cement mortar was carried upwards to the bottom of the foundation at this point, thus completely underpinning and solidly supporting the building foundation and wall.

After this first pier had been allowed to set enough to make certain of a proper support, another pier was excavated for and constructed the same as and about 5 feet from the first one. This process was continued until 20 feet of the walls at both sides of the building were solidly supported upon these brick piers with spaces between them. The spaces between the new brick piers were then excavated to the rock, and brick piers were also built in the openings thus formed, resulting in a continuous new sub-foundation. Then the earth inside the building below the basement floor was protected by a retaining wall at two places, extending across the building and down to the rock, so that the entire space extending clear across the building and some 20 feet lengthwise of the building was excavated down to rock and large heavy immovable boulders, upon which a bed of concrete made a substantial and highly satisfactory bed for the foundation of the new pumping engine. The walls of this pumping station above the main floor had been lined and finished with a high quality of white enameled brick, with very precisely pointed joints; and after the foundation work had been completed, including the foundation for the pumping engine, there was not the slightest trace of any cracks or settlement in the finished

surface of these walls. Such work illustrates the bad policy of loading down an engine contract, as already mentioned in another chapter, with matters which do not belong to it, and which the contractor should not be made responsible for.

Still another, and fully as good an illustration, is a case rather more difficult than the one already cited. A large and important pumping engine, happening to be also of 15,000,000 U. S. gallons daily capacity, although against a considerably greater head, had been contracted for. This engine was also to go into a building already in existence and so situated that it could not be replaced upon the same premises, and therefore it meant the saving of several thousand dollars to the buyer if the existing building could be utilized; only a moderate amount of alterations being necessary in the building itself. The foundations of this building were not any too stable in construction, and rested upon rather a hard limestone bed rock about 12 feet below the main or engine room floor. From the corner of the building the rock surface extended at the depth of 12 feet, about level, in line with one of the walls, and ascended about 4 inches to the foot along the line of the other wall, these two walls forming the 90 degree angle which made the corner of the building.

The outline plans for the new pumping engine showed that a distance of 16 feet would be required from the level of the engine room floor to the under side of the sole plate beneath the water end of the vertical machine; and in addition to this, it was necessary to allow at least 12 inches for concrete in leveling up so as to form a sub-foundation. This meant that the new work would extend down into the rock 5 feet below the building foundation at one side, and 13 feet below the surface of the rock near the middle of the building at the other side of the new engine. The work to be done was to excavate nearly straight down into the rock until a foundation bed was formed, practically although roughly level, and 17 feet below the engine room floor, leaving the foundations of the building along the

walls forming the angle, standing within about a foot of the new excavation in the rock. The building foundation being of not any too compact a character, it was decided to underpin the building with concrete piers extending through the original foundation walls and placed directly beneath the wall piers between the windows, the corner or angle being taken care of in a very complete manner by means of an extra heavy concrete angle pier.

After the walls had been thoroughly protected, the earth was excavated down to the rock, and retaining walls of concrete were formed to hold back the earth in the other parts of the building where no basement existed; and then the earth covering the space where the new engine was to be located, and having dimensions of 42 feet by 24 feet, was excavated and removed from the building, leaving the rock exposed where the new machinery was to be placed. Then commencing near the middle of this area of exposed rock, a pit was formed down into the rock by very light blasting, barring, and picking; this being gradually increased in size and deepened until the required depth was reached, the pit then forming a space at the bottom about 10 feet square. These operations of blasting, barring, and picking were continued around the sides of the pit until the rock excavation was finally completed of the size, shape, and depth. A bed of concrete was then rammed into place until a clean, true, and level surface was formed just the required distance below the engine room floor for the accommodation of the water end of the new pumping engine. The holes for the anchor bolts were then laid out and drilled by diamond core drills to an average depth of about 5 feet, the bolts being secured in the rock by a well known wedging device. (See Fig. 63.)

When an entirely new pumping station is built and arranged for pumping units of uniform size so far at least as floor space is concerned; or where a station already in existence is revamped upon lines of unit capacities, although future units may be of greater capacitiy than the first ones erected, it is

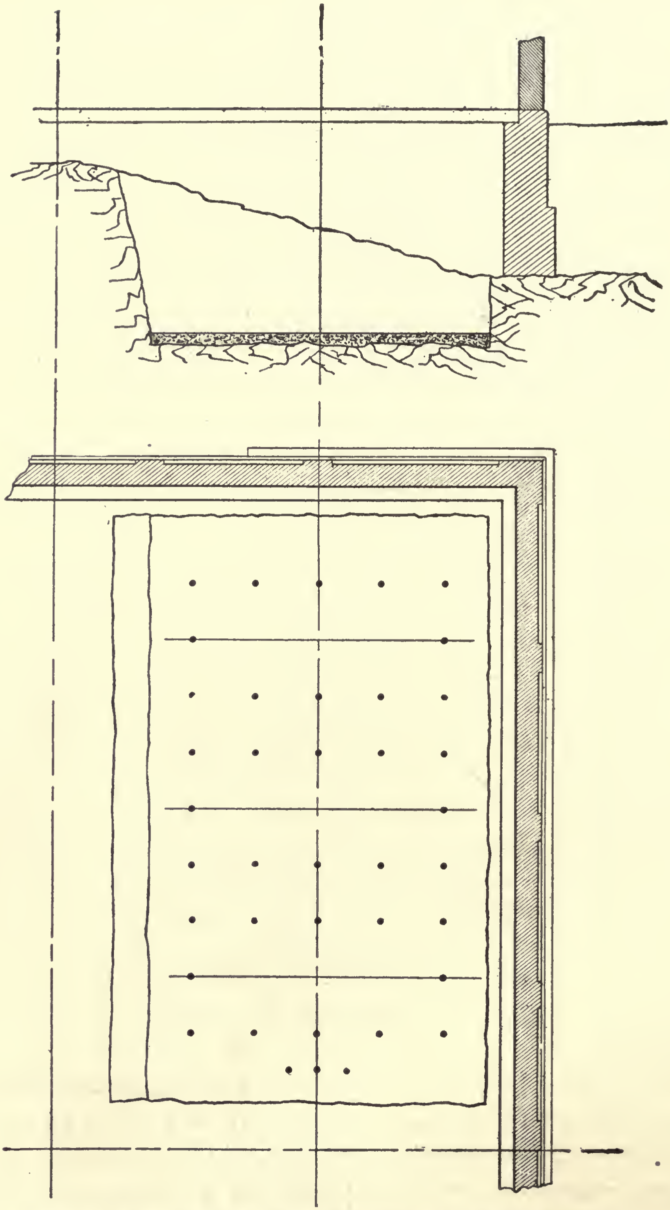


Fig. 63. — Foundations on Rock inside of Pumping Station.

very easy to provide for proper building foundations, engine foundations, pump wells, intakes, screen chambers, etc. But to save unnecessary expense in the future, the arrangements of building and engine foundations, together with the lay out of suction and delivery pipes, must be carefully thought out and planned from the start, even though all of the work is not immediately gone into. There are very few tasks more expensive, annoying, and risky, in proportion to usefulness, than the cutting, carving, and digging necessary for the placing of new pumping machinery and its foundations within a building already crowded with machinery, pipes, and masonry. This of course at times has to be done, and largely no doubt because the additions, improvements, and enlargements are mostly made in established plants possessing limited available space. The matter of economically providing for the future is not so difficult as it may at first seem; and the unit system may be so arranged that a plant, whether large or small, in pumping capacity, could be composed of what might be called compound units, or plant units. Having compound units means to place the machinery in groups, as, for example, four, five, or six pumping engines or pumping units in one complete building, the building forming a plant unit. Then when in the future it becomes apparent that more capacity is needed, instead of attempting to build onto or enlarge an existing building, make the plans for another group or plant unit, conforming of course to any changes or improvements which might have been developed since the last plant unit was built.

In this way, having some definite line to work to the matter of future foundations and other accommodations for the machinery could be intelligently provided for, although no more of the actual work need be done than the present demands call for, the main thing being to look ahead and establish the possibilities for future improvements which, when the time came, could be placed without annoyance, undue expense, or interruptions of the water works service. It will surprise

many to observe, after a little study, how much satisfactory work and planning can be done in this direction.

A great deal could be written upon the subject of pumping engine foundations, but in the absence of opportunities for laying down hard and fast rules which would be of practical use for future work, such writings would naturally have to be confined to work already accomplished; and, beyond the study of materials and methods of construction in particular cases, not much real good could be done. A student, an engineer, or a water works manager would have to determine in the special cases met by them, how far the history or record of work already accomplished could be properly applied to new cases in hand.

Where excavation, building new foundations, retaining walls, underpinning, or similar work is to be done, it will be necessary to ascertain by local conditions and evidence what the character of the earth or soil may be for quite a considerable depth about the locality. In soil, no matter how durable it may appear, even after excavations have been made it will be well to ascertain by drill rods, drills, or augers, or other means of sinking test holes, to what depth a firm and suitable foundation bed may be depended upon. Whenever a considerable depth has to be tested it is very often a good plan to drive down a piece of wrought iron pipe, and, if necessary, this pipe can be made up of moderate lengths successively screwed together by the ordinary pipe couplings so as to increase the convenience of driving. It is not difficult or uncommon in ordinary soil to cheaply go to the depth of 40, 50, or even 100 ft. for testing purposes by the use of the above mentioned or similar appliances.

Rock is of course always to be preferred for the bed of pumping engine foundations, but as it cannot always be reached within a reasonable distance, it is extremely important to determine what course can be safely followed under other and less desirable circumstances. For example, where reasonably tough or strong soil overlies rock to a depth of from 12

to 15 ft. below what would be suitable for the building foundations, but where it would be preferable to have if possible the support of the rock for the engine foundation, it is quite practicable to excavate small circular or square pits through the soil and down to the rock, these pits being dug one at a time and filled with concrete from the rock up to the level of the building foundation, and enough of them put into proper position for taking the support of the engine foundation and thereby transmitting the weight and pressure of the machinery and

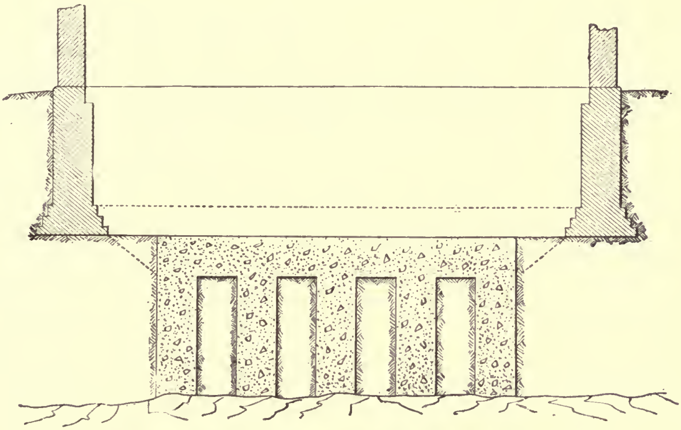


Fig. 64. — Engine Foundations on Concrete Piling.

its foundations directly to the rock without interfering with the stability of the building walls. (See Fig. 64 and Fig. 65.) It is quite practicable to follow this method, even where the soil is not so very firm, by first sinking tubes or caissons made of steel or iron plates, carried down to the rock and afterwards filled with concrete. The result of these methods, either with or without the caisson, is to support the engine foundation upon concrete piles or columns, reinforced by earth.

Such a case is recalled to mind where the pumping station had to be located about 300 ft. from the river edge, and had been built for the accommodation of two horizontal engines with

space left for two more, the engine room being oblong and the engines placed crosswise. The foundations for the machinery had been built upon the same soil as the building walls, a mixture of sand and gravel. The weight of the horizontal engines was quite moderate in proportion to capacity, but rather evenly distributed, and incidentally had the advantage of great area of foundation bottom in proportion to the weight of the machinery, the weight upon the foundation bed not exceeding 400 lbs. to the square foot. This light load had probably not been sought for, but was purely incidental to the shape, dimen-

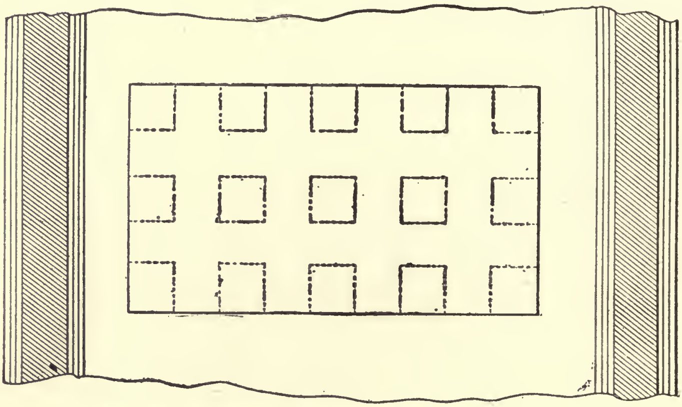


Fig. 65.—Engine Foundations on Concrete Piling.

sions, weight, and disposition of the machinery. These two engines had been in service long enough to have the gross demands for water increase until it very nearly equaled their combined capacity.

After it had been decided to increase the capacity of the station, an investigation of all the circumstances resulted in a recommendation to utilize the remaining space in the building by putting in a vertical, triple expansion, crank and fly-wheel engine, equaling the combined capacity of the two original machines. Fortunately, for architectural effect, the building had been made rather lofty for the accommodation of horizontal

machinery, so that by going down to a considerable depth for foundations, the proposed new vertical engine would just about fit the case. The floor in the reserve portion of the building was removed and excavation commenced. There was a basement to the building, and the foundation of the walls extended a few feet below the basement floor. It was found by sounding and boring that there was bed rock about 20 ft. below the basement floor and practically level beneath the location of the proposed new engine, and it was also found that the upper surface of the lower flat portion of the engine foundations would have to be at the level of the bottom of the building foundations.

It was soon ascertained that the underpinning of the building walls would be pretty expensive, and it was, therefore, decided that 4 ft. would be thickness enough for the bottom portion of the engine foundation, and under the circumstances 4 ft. was considered to be the safe limit of depth to go into the soil below the level of the building foundations. This would leave about 11 or 12 ft. of soil, more or less uncertain in quality and exposure to saturation, between the bottom of the engine foundation and the rock. After looking the matter carefully over, the conclusion was reached that the best plan would be to sink a number of wells, about 4 ft. square, down to the rock and so provide solid concrete supports between the rock and the bottom of the engine foundation. This was successfully carried out, although in some of the holes it was necessary to place sheathings of plank with cross bracing, to safely support the soil while the concrete pillars or piling were being rammed into place. In all cases where the sheathing was used, the bracing was taken out as the level of the concrete was brought up, but in some of the holes it was found necessary to leave the planking in place. The 4 ft. of depth between the top of the concrete piling and the bottom of the building foundations was protected at the sides by planking and bracing until the bed of concrete was safely rammed into place. The planking around these sides having been placed vertically, was easily removed, and the soil was thoroughly compacted against the sides of the

concrete. Upon this platen or cake of concrete, 36 ft. long, 20 ft. broad, and 4 ft. thick, forming a large artificial stone, solidly supported from the rock and with its upper surface at the level of the bottom of the building foundations, the pier for supporting one end of the engine bedplate at the floor level was built. The new engine, of 12,000,000 U. S. gallons capacity against 200 feet head, was erected in place and put into service without any evidence of cracking or settlement of the masonry appearing after the lapse of nearly a year, and it is presumed that the future will show equally as well.

Sometimes situations are met with where a yielding soil is found to overlie firm gravel, hard compact sand, hard pan or rock with the upper material too deep to permit of excavation down to the firm material or to carry down concrete columns as already referred to. In such cases timber piling may be resorted to with success, provided the harder material can be reached by the lower ends of the piling, and through such supports, aided by the compacted earth around the piles, a satisfactory foundation may be secured. But where the soft material is so deep that a hard bottom cannot be reached by the lower ends of the piles nothing but the most urgent necessities should permit the location of pumping machinery in such a place. Of course the compacting of the soft earth between the piles will take place, and a comparatively homogeneous mass will result, but though the individual piles will be little liable to sink, the entire mass as a whole will be likely to settle in the course of time. This may not be particularly injurious to a building or to a dead load, but entails a good many risks to a live moving load, like a pumping or a power engine, risks which had better not be taken. Such things are occasionally done, but they are strongly advised against.

If piling must be resorted to, the situation of the machinery in a particular spot making such a location very important, and the use of piling a secondary matter, especially if a hard bottom can be reached by the lower ends of the piles, these should be placed ordinarily with about $2\frac{1}{4}$ times the diameter

of the pile between centers. The tops should be sawed off to a uniform elevation, and, if necessary, capped with a grill work of timber, some arrangement of heavy planking or other device deemed best for the situation, although too much timber work should be avoided on account of the changes which more or less dryness or dampness of wood may occasion; it being remembered that these changes come laterally, but as a rule not at all lengthwise of the timber. Then a bed of concrete from 3 to 6 ft. in thickness, covering the entire surface of the tops of the piling and the timber, will provide a substantial bed upon which to build the engine foundation. Sometimes where the conditions permit, concrete is placed over and around the heads of the piling without the intervention of the grill work or planking, and, where possible, this method is to be preferred on account of the absence of chances for cross shrinkage of the timbering.

Of course the walls of the building should be firmly supported upon piles, if necessary, but independently of the foundation bed for the machinery. No special rules or directions can be safely given for such work, for the simple reason that when the situation demands such treatment of a foundation, this is evidence enough that each case should be considered by itself. The conditions are too variable and uncertain for hasty conclusions in the absence of a full knowledge of all the facts. The safe load upon piles of 12 inches diameter at the top, when driven in firm soil and without the bottom resting upon hard pan or rock, should not be reckoned at more than 15 to 18 tons per pile, with a dead load, and not more than 8 tons with a live load, like a steam engine at work.

Authorities differ as to the safe load that may be placed upon earth foundation beds, and give all the way from 1 ton to 4 tons per square foot, but mostly still loads are referred to. So far as the writer is concerned however, where permanent and lasting results are looked for, with smooth and economical working pumping engines, the limit is put at 1,000 lbs. per sq. ft., and preferably 800 lbs. for the machinery, and 3,000

lbs. for the building walls of the pumping station, when the bed is the best of earth. An equally distributed pressure is to be carefully sought after and, in the absence of a rock bed or its equivalent, it is well worth the while to pay particular attention to the uniform distribution of the pressure.

As to the matter of material for engine foundations, after the question of bed has been properly decided, a concrete bottom, then the foundation piers of hard burned brick, with granite, limestone, or hard sandstone anchor bolt stones and cap stones will give as good results as any, and better than most other materials or combination of materials. There is a great advantage in brickwork because brickwork can be laid up closely, and true to the required form, with full mortar beds and joints. This material will knit together and make a very manageable and practical construction; and when good hard brick is laid in Portland mortar of proper richness, not too rich, say 1 cement and 2 of sand, there is nothing more to be desired in the way of solidity.

It is not absolutely necessary to use cap stones, although a better piece of work can be done when cap stones are used, but there should be something in the line of washer stones where pockets for the bottom ends of anchor bolts are arranged to be gotten at when desired. But a brick foundation can be built and finished off smooth enough on top for the setting of machinery, if the cap stones are considered as adding too much to the cost of the work. In this case the heavy castings would be leveled up correctly, and then cement and sand grout run under them. A thin mortar of 3 parts cement and 1 part sand will make a very satisfactory grout and it should be made pretty rich with cement to make certain of its running properly under the castings. There should be a liberal space, say 2 inches, between the castings and the brickwork so as to make a solid bed of grout and avoid all chances of the space not filling, as would likely be the case if a thin sheet of grout should be attempted. If the grout is not pretty rich, say 3 cement and 1 sand, it will not flow well, because

the cement will take up and follow the water, while if there be too much sand in the grout, it will settle before the mixture reaches well under a large casting. Probably 4 ft. is about the limit to which grout can be properly run from any one point; so where it is known beforehand that grout is to be used, provisions should be made for pouring from several places about the casting. The writer observed recently where some large horizontal engines were being bedded, on top of brick foundations, that rather heavy cement mortar was being rammed and packed beneath the castings by means of wooden paddles, the castings being supported the meanwhile upon wedges; the space underneath these castings was about $1\frac{1}{2}$ inches and the work seemed to be progressing in a very satisfactory manner.

The matter of concrete is a very important one, and a strong compact mixture should be made for engine foundations, whether the entire foundation is made of concrete, as is sometimes done, or whether the lower part or bed only is formed of this material. The best working mixture for concrete for foundations or for work about water, such as core walls, and similar work pertaining to dams, reservoirs, gate houses, etc., is in the writer's opinion 1 part Portland cement, 2 parts sand, and 4 parts crushed stone. Such a mixture will fill the voids completely and make a close grained, solid artificial stone as may be seen from the following :

If a wooden box is made, of one cubic yard in measurement, or 36 inches long, broad, and deep, its cubical contents will be $36 \times 36 \times 36$, equal to 46,656 cubic inches. And if a perfect globe of stone with a diameter of 36 inches is placed in such a box, of course it would touch the top, bottom, and the four sides. The cubical contents of the globe would be the cube of the diameter multiplied by the decimal 0.5236, or it would be $36 \times 36 \times 36 \times 0.5236$, equal to 24,429 cubic inches. The empty space will then be the difference between these two quantities, or nearly 48 per cent or say practically one half solid and one half space or void.

If globes 3 inches in diameter were placed in the box, there will be 12 globes each way for 12 layers, or 12 × 12 × 12, equal to 1,728 of the 3 inch globes. The cubical contents of one of the 3 inch globes would be 3 × 3 × 3 × 0.5236 or 14.1372 cubic inches, which multiplied by the number of 3 inch globes, 1,728, will give 24,429 cubic inches or the same as with the single globe of 36 inches diameter. This of course means that, whatever the size of the globes, the result will be that there will be practically half solid and half space or voids.

Crushed stone is not as regular as the globes just referred to, but the irregularities amount to about the same thing, fitting together in some places and holding apart in others, so that take it altogether the rule of half solids and half voids will hold good. Therefore when the cubic yard box is filled with crushed stone, there will be room for half a cubic yard of sand in the spaces between the stones; and again, after the sand is in place, there will be room for half as much cement in the spaces between the sand grains, or a quarter of a cubic yard for cement. As 8 ordinary cement barrels full of crushed stone will make a cubic yard, there will be 4 barrels of sand and 2 barrels of cement in the mixture, or 2 cement, 4 sand, and 8 stone; or, as at first given, 1-2-4 for the best and strongest mixture for concrete. This cannot be carried out as exactly as the globes indicate, but the barrel measurement given above will answer all practical purposes.

To show how closely the sand will follow the same law as the crushed stone, it may be assumed for the purpose of calculation that the sand grains are globes one fiftieth, 0.02, of an inch in diameter; then 0.02 × 0.02 × 0.02 × 0.5236 will equal 0.0000041888 of a cubic inch as the cubical contents of a grain of sand. At one fiftieth of an inch diameter there will be 1,800 grains each way in the 36 inch box, which will make 1,800 × 1,800 × 1,800 or 5,832,000,000, which multiplied by the cubical contents of each grain will give 24,429 cubic inches, as before, as the solid contents of a cubic yard of sand, leaving about half a cubic yard as voids if the box contained only sand.

Engine foundations are sometimes built of stone, but unless of cut stone or at least of dressed stone with true beds and joints, this material is nothing like so good as brick. Rubble stone for such work is very objectionable, unless a good deal of dressing of the stone is done, and this would not pay in the absence of very cheap material. Rough, thick beds and joints in a foundation for supporting a live load are not to be commended at all; they are likely to lead to trouble with the machinery. Rubble work when put in with reference to core walls and like work, where it can be made water tight but practically supporting no load to speak of, is very acceptable; but for pumping engine foundations it should be avoided.

Concrete foundations are coming more and more into use, and some very well appearing work has been done in this line, although for general work, if the truth were known, when proper materials and mixtures are employed, the cost of forms is considered, etc., the price per cubic yard will go above brickwork trimmed with stone, and would be a useless expense. There has been a sort of fad on concrete of late years, but it will eventually settle into its proper place and like all other good things play its allotted part of usefulness. It is excellent for subfoundations where no special or exact form is needed, and where it can be rammed into place and brought up to some general level, to build the regular foundation upon; but is not desirable for the upper work in important plants unless special reasons exist for its use.

Large, vertical, self contained pumping engines resting upon a bottom or sole plate, extending beneath the entire machine, need nothing more than a flat layer of concrete underneath, although the concrete should be bedded on rock or have some equivalent support; and in such a situation there is probably nothing better than concrete, and considering the difficulties often met with in the lower parts of such work, possibly it is the least costly construction that can be employed.

CHAPTER XVI

INVESTMENT VALUE OF PUMPING ENGINES

These tables exhibiting the investment value of pumping plants containing various types and classes of pumping engines with adequate boilers, are based upon data which are considered to be such as will show practical conditions in the average run of plants in actual use in water works service. Of course it is impossible to make a schedule which will hit all plants; but the water pressure given will be found to be a safe basis for calculations, and although a little higher or a little lower pressure will vary the actual figures for power and fuel, the relations will not change much under the same conditions. It will be safer to consider any particular case by itself, but the tables will indicate closely approximate results. The data of the tables are as follows:

Number of days for a year's work	365
Number of hours of pumping each day	16
Number of watches each day	2
Length of watches in hours	8
Pay of operating engineers per year	$1,200
Pay of firemen per year	600
Pay of extra men per year	600
Maintenance account of engines	3%
Interest account of engines	4%
Sinking fund for vertical triple engines	3%
Sinking fund for all other types of engines	5%
Oil, waste, packing, and small repairs	1%

The calculations in the following tables are based upon a fair average price for machinery, foundations, and appurtenances, together with boilers and their appliances. Also upon an actual evaporation in the boilers of 8 lbs. of steam produced at the working pressure with one pound of coal; also

upon a price paid for the coal of $3 per net ton of 2,000 lbs. in the fire room ready for firing; also upon a total water load upon the plungers of 90 lbs. per square inch, or a total working head of 207 ft., including suction and friction of the water.

The desire is frequently expressed by interested parties; for a sort of schedule or rate of cost or price of pumping engines; but it is a very difficult matter to make a list of such costs which at any certain time will be reliable beyond an approximate guide for estimation. In fact all prices pertaining to specially defined contracts are more or less fickle and changeable. About January, 1899, prices of all sorts of materials began to rise, and by 1901 were at a point higher than for several years. In the spring of 1904, a downward tendency developed and coke ovens in western Pennsylvania were shut down, pig iron began to decline, Portland cement fell decidedly off in price, steel products were lower, cast iron water pipe was at $23 per ton, and a general fall in prices threatened. But by the autumn of the same year (1904) the drooping markets again strengthened and have advanced steadily since, until now, January, 1907, cast iron pipe is above $30 per ton, steel products are high and hard to get, and orders are booked for a year ahead.

In 1900 the city of Cleveland rejected all bids for pumping machinery at an attempted letting at that time, because the prices asked were so high as to seem prohibitive; and the needed machinery was bought two years later at better figures for the city.

Based upon the low figures of 1896, the writer has a record of bids on pumping engines per million gallons, varying from the bottom upwards, as follows:

A. Bottom figures of 1896, represented by	100%
B. Later figures	137%
C. Still later figures	118%
D. Comparatively recent figures	155%

So it will be observed that about the best attempt can only result in a fair average "subject to change without notice,"

according to the labor and material markets, and also depending upon the state of the shops bidding upon the special work in question. When a shop is "hungry for work" bids will be low; and when the shop is "full up with work" bids will be high. All shops do not strike the tide at the same stage at the same time, and hence a certain amount of "irregularity" in the market with a strong or a weak undertone as the case may be according to the time.

A fairly good guide for complete plants is given herewith which is sufficiently close for preliminary estimates, but conditions should be looked into carefully for exact estimates. In some actual cases these figures may be a little too high and in other cases too low; but they are closely approximate and enough of them are based upon records to fairly insure the table as safe for practical use. In fact, the table is so close that it would be taking chances for a contractor to guarantee the production of results for the figures named, without investigating each case by itself. The work contemplated is for the best type of triple expansion pumping engines and high pressure boilers, a good design and quality of brick buildings, or of stone where stone is cheap, steel trussed and slate covered roofs, adequate chimneys, properly proportioned and thoroughly screened intakes. The cost given includes everything excepting the land.

Cost of complete pumping stations.

Pounds Pressure per Square Inch of the Water Load Worked Against.	Cost per Million Gallons of Plant Capacity, Including a Reserve.
30	$ 6,750
40	7,000
50	7,250
60	7,500
70	7,750
80	8,000
90	8,250
100	8,500
110	8,750
120	9,000
130	10,000

There are cheaper types of pumping engines, but they are necessarily of lower efficiency, and therefore requiring more boiler capacity, more coal storage, and other incidentals which, when balanced up, will tend to keep the figures about the same. A cheaper and less durable sort of building may be used, but in the long run this will call for more repairs, which when capitalized will bring the account fully back to the above figures, and most likely exceed them.

The following series of tables show the cost of installation, maintenance, operation, repairs, etc., of pumping engines complete with piping, foundations, appurtenances, etc., within the engine room ready for continuous operation in service. Also the cost of the necessary boiler plant within the boiler house, including stokers, setting, feed pumps, all piping and appurtenances complete. The figures do not include anything for buildings, land, chimneys, wells, etc.

Cost of pumping engines complete, with foundations, piping, and appurtenances, per million gallons per 24 hours capacity.

Compound condensing, low duty	$2,300
Low duty triple condensing	2,800
High duty, compound horizontal, condensing	3,300
High duty, vertical triple, condensing	4,800

Cost of boilers with stokers, setting, feed pumps, and appurtenances complete.

Per boiler horse power	$20
Maintenance account of boilers	5%
Interest account of boilers	4%
Sinking fund of boilers	5%

Table showing the capacity per 24 hours, duty per 1,000 lbs. of steam, steam consumed per hour, number of running engineers, firemen, and extra men, and boiler horse power.

Million Gallons Capacity per 24 Hours.	Duty in Foot Pounds per 1,000 Pounds of Steam.	Pounds of Steam Consumed per Hour.	Number of Running Engineers.	Firemen.	Extra Men.	Required Boiler Horse Power.
Compound Condensing Low Duty Pumping Engines.						
3,000,000	50,000,000	4,276	2	2	0	150
4,000,000	55,000,000	5,184	2	2	0	175
5,000,000	60,000,000	5,940	2	2	0	200
6,000,000	60,000,000	7,128	2	2	0	250
7,000,000	65,000,000	7,676	2	2	0	275
8,000,000	65,000,000	8,772	2	2	0	300
10,000,000	65,000,000	10,966	2	2	0	375
12,000,000	70,000,000	12,269	2	4	0	400
15,000,000	70,000,000	15,336	2	4	0	500
Triple Condensing Low Duty Pumping Engines.						
3,000,000	75,000,000	2,851	2	2	0	100
4,000,000	80,000,000	3,564	2	2	0	125
5,000,000	85,000,000	4,194	2	2	0	150
6,000,000	95,000,000	4,752	2	2	0	175
Compound Condensing High Duty Pumping Engines.						
5,000,000	110,000,000	3,240	2	2	0	100
6,000,000	112,000,000	3,819	2	2	0	125
7,000,000	114,000,000	4,337	2	2	0	150
8,000,000	116,000,000	4,916	2	2	0	175
10,000,000	119,000,000	5,990	2	2	0	200
12,000,000	122,000,000	7,011	2	2	0	250
15,000,000	125,000,000	8,554	2	2	0	300
Triple Condensing High Duty Pumping Engines.						
6,000,000	140,000,000	3,050	2	2	0	100
7,000,000	145,000,000	3,442	2	2	0	125
8,000,000	150,000,000	3,802	2	2	0	125
10,000,000	155,000,000	4,597	2	2	0	150
12,000,000	160,000,000	5,348	2	2	0	175
15,000,000	165,000,000	6,480	2	2	0	200
20,000,000	170,000,000	8,388	2	2	2	275
25,000,000	175,000,000	10,179	2	2	2	350
30,000,000	180,000,000	11,880	2	4	2	400

Cost of pumping plants including pumping engines and boilers, foundations, piping, appurtenances, etc. Complete ready for service.

Million Gallons per 24 Hours.	Pumping Machinery, Foundations and Piping.	Boilers, Setting, Piping and Appurtenances.	Total Cost.
Compound Condensing Low Duty Pumping Engines.			
3,000,000	$ 6,900	$ 3,000	$ 9,900
4,000,000	9,200	3,500	12,700
5,000,000	11,500	4,000	15,500
6,000,000	13,800	5,000	18,800
7,000,000	16,100	5,500	21,600
8,000,000	18,400	6,000	24,400
10,000,000	23,000	7,500	30,500
12,000,000	27,600	8,000	35,600
15,000,000	34,500	10,000	44,500
Triple Condensing Low Duty Pumping Engines.			
3,000,000	$ 8,400	$2,000	$10,400
4,000,000	11,200	2,400	13,700
5,000,000	14,000	3,000	17,000
6,000,000	16,800	3,500	20,300
Compound Condensing High Duty Pumping Engines.			
5,000,000	$16,500	$2,000	$18,500
6,000,000	19,800	2,500	22,300
7,000,000	23,100	3,000	26,100
8,000,000	26,400	3,500	29,900
10,000,000	33,000	4,000	37,000
12,000,000	39,600	5,000	44,600
15,000,000	49,500	6,000	55,500
Triple Condensing High Duty Pumping Engines.			
6,000,000	$ 28,800	$2,000	$ 30,800
7,000,000	33,600	2,500	36,100
8,000,000	38,400	2,500	40,900
10,000,000	48,000	3,000	51,000
12,000,000	57,600	3,500	61,100
15,000,000	72,000	4,000	76,000
20,000,000	96,000	5,500	101,500
25,000,000	120,000	7,000	127,000
30,000,000	144,000	8,000	152,000

Table showing cost of coal per year for 16 hours pumping per day, maintenance, interest, sinking fund, oil, waste, packing, small repairs, etc., for the pumping engines, wages per year for operating the boilers and pumping engines, and the sinking fund, maintenance, and interest of the boiler plant. The last or right hand column shows the cost of pumping per million gallons.

Capacity of Pumping Engines per 24 Hours in U. S. Gallons.	Cost of Coal per Year of 365 Days.	Maintenance, Interest, Sinking Fund, Oil, Waste, Packing, Etc.	Wages per Year.	Maintenance, Interest, Sinking Fund, of Boilers.	Cost of Pumping per Million Gallons.
Compound Condensing Low Duty Pumping Engines.					
3,000,000	$4,687	$ 897	$3,600	$ 420	$13.15
4,000,000	5,672	1,196	3,600	490	11.26
5,000,000	6,504	1,495	3,600	560	9.99
6,000,000	7,807	1,794	3,600	700	9.52
7,000,000	8,410	2,093	3,600	770	8.93
8,000,000	9,600	2,392	3,600	840	8.44
10,000,000	11,913	2,990	3,600	1,050	8.07
12,000,000	13,320	3,588	4,800	1,120	7.81
15,000,000	16,784	4,485	4,800	1,400	7.75
Triple Condensing Low Duty Pumping Engines.					
3,000,000	$3,118	$1,092	$3,600	$280	$11.08
4,000,000	3,907	1,456	3,600	350	9.57
5,000,000	4,590	1,820	3,600	420	8.57
6,000,000	5,203	2,184	3,600	490	7.85
Compound Condensing High Duty Pumping Engines.					
5,000,000	$3,556	$2,145	$3,600	$280	$7.85
6,000,000	4,161	2,574	3,600	350	7.31
7,000,000	4,800	3,003	3,600	420	6.94
8,000,000	5,379	3,432	3,600	490	6.62
10,000,000	6,550	4,290	3,600	560	6.19
12,000,000	7,665	5,148	3,600	700	5.86
15,000,000	9,355	6,435	3,600	840	5.75
Triple Condensing High Duty Pumping Engines.					
6,000,000	$3,285	$3,168	$3,600	$280	$7.07
7,000,000	3,766	3,696	3,600	350	6.70
8,000,000	4,161	4,224	3,600	350	6.33
10,000,000	5,028	5,280	3,600	420	5.91
12,000,000	6,025	6,336	3,600	490	5.64
15,000,000	7,096	7,920	3,600	560	5.45
20,000,000	9,180	10,560	4,800	770	5.22
25,000,000	11,142	13,200	4,800	980	4.95
30,000,000	12,999	15,840	6,000	1,120	4.92

Table showing opposite the rated capacity of the different engines, the gallons which would be pumped per year at full speed and constant pumpage. Also showing the gallons per year with a pumpage of 16 hours per day. The latter rate taken as a fairly good average for water works plants. The figures show U. S. gallons.

Capacity of the Pumping Engines per 24 Hours.	Capacity of the Pumping Engines per Year Working 24 Hours per Day.	Capacity of the Pumping Engines per Year, Working 16 Hours per Day.
3,000,000	1,095,000,000	730,000,000
4,000,000	1,460,000,000	973,000,000
5,000,000	1,825,000,000	1,217,000,000
6,000,000	2,190,000,000	1,460,000,000
7,000,000	2,555,000,000	1,703,000,000
8,000,000	2,920,000,000	1,947,000,000
10,000,000	3,650,000,000	2,423,000,000
12,000,000	4,380,000,000	2,920,000,000
15,000,000	5,275,000,000	3,517,000,000
20,000,000	7,300,000,000	4,868,000,000
25,000,000	9,125,000,000	6,083,000,000
30,000,000	10,950,000,000	7,300,000,000

The first thing noticeable in these tables is that the cost of pumping per million gallons, all expenses included, gradually decreases from the smallest engine of the low duty compound condensing class, to the largest of the high duty triple expansion class. And this decrease is brought about partly by the gradual increase in economic duty and partly by the increase in capacity; some of the charges per million gallons being directly affected by the increase in capacity, as, for example, a 5,000,000 gallon engine requires practically as much per annum for the wages of the men as a 10,000,000 gallon engine; and, of course, twice the water is pumped with the latter as with the former which cuts the expense for wages in halves, per million gallons. And, further, a higher duty reduces the coal account and the fixed charges against the boiler plant per million gallons.

But the cost per million gallons does not tell all of the story. Because the total cost in money for pumping all of the water per annum, although lower per million gallons, may not be enough less *total* per annum, with an engine of higher duty and of higher first cost to justify the extra investment. Plants

equally constructed and with equally low repair and maintenance accounts, can have no advantage over each other excepting in the matter of fuel economy; and the lessened amount of coal to be bought is the real foundation upon which to base an increased investment in the plant. And, therefore, in considering this, it will be perceived, as intimated above, that there are only two items which grow less by higher duty, and these are the coal account and the fixed charges on the boiler plant. Everything else increases with higher duty excepting the wages account, for equal capacities, and this account remains at least as much with high as with low duty, excepting with the very large high duty triple engines, and with these the wages are somewhat less in the fire room on account of the lessened amount of coal to be shoveled in proportion to the pumping.

The items for and against the high duty account may be tabulated as follows:

Against High Duty.	*In Favor of High Duty.*
Maintenance account for machinery.	Maintenance account for boilers.
Interest on machinery.	Interest on boilers.
Sinking fund for machinery.	Sinking fund for boilers.
Oil, waste, packing, etc.	The coal account.

The making of comparisons for the purpose of ascertaining before machinery and boilers are contracted for, which type and class it will pay best to buy, requires comparisons and calculations which call for some thought and care, but which may be readily enough made when the actual conditions of the contemplated plant are clearly laid down. Of course it goes without saying that the lowest cost or investment in a pumping plant is the most desirable, all other things being at least equal, or at least not against the lower cost plant. This is dictated by the most ordinary laws of nature and common sense, which insistently call for economy in all things; but this same natural and insistent demand for economy is the very spur which urges us onward towards a greater investment in a pumping plant as well; but in the case of the plant of the same capacity at a higher cost, the economy takes the form of economy in opera-

tion instead of economy in first cost; the total sum of all expenses being less with the plant of higher first cost. Therefore, the lowest investment or cost for the plant being the first thing to consider, and the fact that valuable inducements must be offered to bring a higher cost plant into favor, it follows that the low duty engines must be the basis for comparison, and the starting point for more desirable developments.

So far as these tables are concerned, and the conditions, values, pressures, powers, and quantities which they set forth, any plant costing more than the regular low duty, compound condensing plants here shown, must show itself capable of saving enough money by economizing coal to pay the different items enumerated, to the extent represented by the increased cost of the higher duty and higher priced plant.

To illustrate this idea it may be noted that the fixed charges against the machinery and boilers are as follows:

Vertical, Triple Expansion Pumping Engines.

Maintenance account	3%
Interest account	4%
Sinking fund account	3%
Oil, waste, packing, and small repairs	1%
Total fixed charges	11%

All other types of Pumping Engines.

Maintenance account	3%
Interest account	4%
Sinking fund account	5%
Oil, waste, packing, and small repairs	1%
Total fixed charges	13%

NOTE. — The reason that there is a difference of 2 per cent made in the sinking fund account, between vertical triple machinery and all other forms, is that it has become evident that the vertical triple type of pumping engine represents just about the limit of possible steam economy, and therefore the vertical triple type will probably not be replaced as obsolete by any of the other types in the tables, but the other types will eventually give way to the vertical triple machinery. Under these circumstances the life of the vertical triple when properly built is taken at 33⅓ years, and that of all other of the types given at 20 years; representing, respectively, 3 per cent and 5 per cent to make up the sinking fund to 100 per cent in the times specified.

The fixed charges against the boiler plant are as follows:

Maintenance account	5%
Interest account	4%
Sinking fund account	5%
Total fixed charges	14%

To make an application of the tables to an actual case, let us take for example a 6,000,000 gallon plant, as that capacity appears in all of the types and classes.

The total cost of the different plants, including boilers, appears as follows:

Low duty compound	$18,800
Low duty triple	20,300
High duty compound	22,300
High duty triple	30,800

The total cost for operating per annum appears as follows:

Low duty compound	$13,902
Low duty triple	11,477
High duty compound	10,685
High duty triple	10,333

Beginning the comparison from the low duty compound, up to the low duty triple, it is found that the difference in yearly cost of owning and operating is 2,425 in favor of the low duty triple; and as the difference in cost of plant is $1,500, and 13 per cent (the fixed charge) of this difference amounts to $195, it is clear that a saving of $2,425 per annum will justify the use of a low duty triple against a low duty compound plant.

Now going still further up the line, and figuring upon a high duty compound, it will be found that the difference in yearly cost between the low duty triple and the high duty compound amounts to $720 in favor of the high duty compound; and as the difference in cost of plant is $2,000, and as 13 per cent (the fixed charge) of this latter amount is $260, there is a balance of $502 per annum in favor of the high duty compound.

In making the comparison directly between the low duty compound and the high duty compound, the difference in cost

per annum and its commercial effect is still more striking. The difference in cost between these two latter plants amounts to $3,500 against the high duty compound, but the difference in annual expenses amounts to $3,217 in favor of the high duty. Then as the fixed charge of 13 per cent on the difference in cost of plant is only $455 there is a profit of $1,762 per annum between low and high duty compound plants.

Going still further up, to the high duty triple, it will be found that the difference in yearly cost between the high duty compound and the high duty triple amounts to $352 in favor of the high duty triple. The difference in cost of plants is $8,500, but as the 11 per cent (the fixed charges for high duty triples) amounts to $935, or $583 against the high duty triple per annum, it appears that with 6,000,000 gallon engines against 90 lbs. water load, and with coal at $3 per net ton in the fire room, the high duty compound will be the most profitable plant to buy and use.

The high duty triple would have only $11 per annum against it when compared with the low duty triple; and the high duty triple would have $2,249 in its favor per annum against the low duty compound, with the fixed charges of 11 per cent against the high duty triple.

Therefore, under the conditions given in the tables, the 6,000,000 gallon high duty triple with its boilers would be much preferred to the low duty compound; would be just about equal as an investment to the low duty triple; and would not be so good an investment as the high duty compound.

A reduction in the price of coal, but with the coal equal in quality to the $3 fuel, the advantage would change towards the lower classes of engines, and with the coal at a higher price than $3 the changes would be in favor of the higher classes of engines.

The low duty compound pumping engines above 2,000,000 to 3,000,000 gallons daily capacity have been practically retired from the water works field, or should be so far as a desirable investment is concerned, and the introduction of the low duty

triple has even made the small compounds of extremely doubtful utility when any reasonable economy of investment and operation is looked for. The low duty triple in its turn is restricted by the high duty compound at the 5,000,000 or 6,000,000 gallon capacity, and therefore the low duty triple is limited at 6,000,000 in the tables. The high duty compound disputes the field with the high duty triple from 6,000,000 gallons to 10,000,000 gallons capacity, according to the cost of the fuel used mostly, but also on account of special conditions sometimes. Above the 10,000,000 gallon daily capacity it requires especially adverse conditions to bar out the vertical triple engine as the best investment in machinery for pumping water, all things considered.

After the 10,000,000 gallon mark is passed the high duty compound commences to meet with difficulties of construction with reference to making a reliable and durable machine within costs which do not approach those of the triple too closely to justify its use. Also, the question of extremes in low and high water involving suction lift and flood water, in many places, argue against large horizontal pumping engines, it being remembered that the suction lift measures up to the discharge valve decks and these become pretty high above the floor line in good sized machines; also the matter of more expensive building comes in for consideration, giving a basement low enough to enable the engine to reach the water easily at low stage, which is by far the most prevalent stage, and also to have the building so constructed that the machinery will be protected from the flood water when the level rises. It does no special harm to flood the water ends of vertical engines so long as the wheels are above the water, but the horizontal machine is brought to a standstill when the water reaches the steam cylinders and moving parts.

The table of high duty compounds is carried up to 15,000,000 gallon engines for the purpose of making comparisons, but as the vertical triple machine has been brought to a commercial basis practically, the vertical idea in the compound involv-

ing as it does more or less special construction, places the compound at a disadvantage as to cost. The cost of high duty compounds given in the tables refers to horizontal machines, and as there are only two standard engines of this type, the Worthington and the Gaskill, that can be brought within these figures unless we go to cross compounds, which have objections in the large sizes, and further have not been brought to a commercial basis to the degree represented by the two above mentioned. There have not been many pumping engines of the horizontal type above 10,000,000 gallons capacity built in recent years, and it looks as though there would be very few, if any, so large built in the future. With coal at $3 per ton, the horizontal compound high duty engine can hold its own as an all around investment against the vertical triple, but the feeling against the horizontal class, with comparatively short strokes, and with very large steam cylinders placed in a horizontal position, will no doubt bar them out of the larger plants.

Other forms of heat engines for water works, in the form of the steam turbine pumping engine, which would likely consist of a steam turbine engine coupled directly to a turbine pump, and also the gas engine as a medium for pumping water, begin to loom up ahead. But nothing definite has yet been accomplished upon a scale large enough to command respect. The turbine has a very doubtful future before it, as the promises involved in the gas engine question make the latter a very formidable competitor at some time in the not very distant future, even for the present record holders.

CHAPTER XVII

SUCTION LIFT AND SUCTION PIPES

The water end, or the main pumps of a water works pumping engine and the proper and efficient action of this portion of the machine, are the principal reasons for the existence of the apparatus. The operation of pumping water is extremely simpel, but there are a few, very few, cardinal or main details which must be strictly observed if success is to attend the efforts of the pump maker. It is pretty safe to say that, aside from sufficiently strong construction, the most important detail in the operation of pumping is to get the water into the pump through its suction valves; and for the simple reason that the force or pressure available for that part of the work comprised in what is called "suction lift" and the complete filling of the plunger chambers, is very limited indeed, in fact, limited to the comparatively slight pressure of the atmospheric air which we breathe in our lungs. A great abundance of force is available for discharge, or forcing the water out of the pump, but the height to which the water may be "lifted" above the surface of the well from which it is drawn, and the velocity or speed which it can be made to take through the suction pipes and pump valves, are absolutely fixed and moderated by a very limited amount of force.

Under normal conditions and with the pump empty of water and at rest, the water stands at the level of the well supply, and the natural air is in the pump and well alike. But the moment an attempt is made to displace the air within the pump chambers and suction pipe, by the motion of the plungers, or by any other means such as driving the air out of the pump by means of a steam or water jet, or by any means whatever,

then the natural pressure of the air within the pump becomes less than the pressure of the outer air having access to the pump well, just in proportion as the removal of the air within the pump is carried to completion.

Under the ordinary conditions of the atmosphere we breathe and when the temperature is moderate, say from 40 to 90 degrees, according to the ordinary thermometer we are accustomed to see every day, the pressure of the air is about 15 lbs. per square inch; to be exact, with the air at 60 degrees temperature at sea level the pressure is 14.7 lbs. And, if all of the air is removed by any means from the inside of an air tight vessel, as, for example, the pipes and chambers of a pump, and so that the pump is completely emptied of air, then the outside air will exert a force or pressure of about 15 lbs. to the square inch upon all of the outside surface of the pump or vessel so emptied, and there will be no corresponding or balancing pressure inside of the pump or other vessel. Therefore, it follows, that if this outer air is pressing down upon the surface of the water in a pump well, which it is at all times, if the air is removed from the inside of the pump, and if a pipe connects from beneath the surface of the water in the well to the interior of the pump, then the water in the well will be forced up and through the pipe and into the pump by the atmospheric or natural air pressure at the outside.

This action is perfectly satisfactory and complete so far as it is able to go. But, as already pointed out, the weight, force, or pressure of the natural air being fixed and limited within certain narrow bounds, and the weight of water being also fixed and limited in its weight within certain narrow bounds, it follows that there is also a limit to the height to which the air pressure will force or "lift" the water in a pipe or vessel. And it is within this limit of height or lift that we must manage to keep when attempting to successfully design or operate a pumping engine. This action of the natural air lifting or forcing water up a pipe or tube when the air within the pump or other vessel is removed or partially removed, is technically called

and is generally known by the name of "suction." And the word "suction" is good enough, perhaps, for the purpose of expressing a certain action or performance, but in the sense of its meaning that the water is drawn up the pipe and into the pump, it is absolutely in error and out of place. In fact there is no such action as "suction" so far as it is made to mean the "pulling" or "drawing" of water from a lower to a higher level by a force situated higher up or at a higher level. The water is not drawn up at all by something in front of it. It is driven up or forced up by something behind it. And that something behind the water is the natural atmosphere or air we breathe; and speaking of breathing, the action of a pump is of precisely the same nature as breathing; the muscles of the chest and pulmonary regions distend or enlarge the chambers and passageways in the lungs, thus causing a partial vacuum, and thus enabling the outer air to force its way into what we call our breathing spaces. To illustrate this, let any one place the tube of an ordinary vacuum gauge to the lips, close the nose and then go through the operation of breathing or drawing in breath. The vacuum gauge will at once indicate to what extent such muscular efforts are effective. A vacuum from 18 to 21 inches on a mercury column, or a Bourdon tube gauge, may be readily formed by the breathing experiment. The attempt to form a vacuum in a pump will be followed by precisely the same result; only a body of water being situated between the outer air and the inner space, instead of the air getting in as in the case of breathing with the lungs, the outer air drives the water into the pump ahead of itself, the result being that so long as water enough comes into the well to keep the water surface above the bottom of the pipe leading into the pump, commonly called the suction pipe, no air can get in at all, but a constant stream or flow of water will take place up the pipe and into the pump. The term "vacuum," as many people are more or less aware, signifies a complete emptiness and absence of everything, air, vapor, gas, water, and all else that we know of. And into a vacuous space any and all gases,

atmospheres, or any expansive fluid will endeavor to penetrate and flow. A vacuum has no substance or properties of its own, beyond the property of complete emptiness, and does not draw nor suck nor do any kind of work. It is simply a condition which offers a chance for work to be done.

The extreme height to which the natural air pressure will force or drive up a column of water into a vacuum, is about 34 ft., which simply means that within a vessel containing no air whatever nor anything else, but connected by a pipe with a well containing water, a column, or head, or height, of water of about 34 ft. will just balance the entire pressure or weight of the natural air resting upon the surface of the water in the well. The water will go no higher than about 34 ft. no matter how perfect the vacuum may be, or how large the chamber or vessel may be above the 34 ft. level. This head of 34 ft. of water represents a pressure of about 15 lbs. per square inch, or, to be more exact, a pressure of 14.7 lbs. per square inch.

Considering the fact that the getting of the water up the supply pipe and into the pump is widely recognized by the technical name of "suction" it will be expedient and convenient to preserve the expression and so this part of the operation of a pump will be so designated, the foregoing explanation that there is no such thing as a pulling action in the suction process having been made simply to place the actual facts in the case clearly before the mind of the reader and student. The practical effect then of suction in a pump is to utilize a portion of the 34 ft. head, or say 14.7 lbs. of force to overcome the difference in level between the surface of the well and the height in the pump to which the water must be "lifted" by suction; and then utilize the balance or remaining portion of this force for maintaining the flow of the water into the pump by overcoming the friction and other resistances in the moving water. And in following out this idea, it becomes at once plain that the higher the water must be lifted into the pump, the slower must be the rate of flow, as the available force of 14.7 lbs. is, within very narrow limits, a fixed and

unalterable one; the logic of this being that with no lift at all to the water, the entire force of the atmosphere, 14.7 lbs., would be available for overcoming the friction of flow; and contrary to this, with a lift equal to 34 ft. or the equivalent of the entire atmospheric force of 14.7 lbs. there would be nothing at all available for overcoming the friction of flow through the suction pipe and into the pump.

Therefore, when there is a choice in the matter as to how much to allow for lift and how much to allow for flow, judgment based upon experience comes into the case, and good practice, when free to choose, chooses from 12 to 15 ft. for the lift when the well is fairly close to the pumps and is supplied with water by an easy flow from the source of supply; and the balance of from 19 to 22 ft. is then available for overcoming the resistances to the flow through the suction pipe and into the plunger chambers. A suction lift of 20 ft. could be tolerated and sometimes we are confronted with conditions demanding more than 20 ft. lift, even so high as 26 ft., but every possible effort should be made to avoid going above 20 ft., and if a height of so much as 26 ft. is absolutely unavoidable under any certain set of conditions, then of course it must be met, but it will be with a feeling that the best and most desirable results in pumping water are unattainable. Where it is necessary to flow the water through an artificial conduit for a considerable distance and into a limited well for the engine to take its supply from, a suction lift of 26 ft. would be absolutely perilous to the machinery, and in such a case the level of the water in the well should be kept as near to the elevation of the discharge valve decks as possible, say within 4 or 5 ft.

Even the actual difference between the surface of the water in the well and the extreme height of necessary suction limited to 20 ft., there will then be a loss in getting the water through the suction valves of the pump before the net amount of pressure available for flow becomes apparent. The resistance of the water passing through a set of suction valves properly proportioned to a pump, is taken as the results of experiment

and observation at one foot head; so that we have the summary of the suction or inlet account as follows for good economical limits with a liberal well close at hand, and a free flow from the source of supply:

Actual net lift accomplished between the level of the water in the well and the under side of the discharge valves in the main pumps in the engine . .	20 feet.
Resistance of the suction valves	1 foot.
Available for friction of flow	13 feet.
Total atmospheric force at sea level.	34 feet.

The unemployed head of 13 ft., aside from the height of the water column and the resistance of the suction valves and seats, is not all available for overcoming friction in a straight pipe and so it is not safe to work completely up to the limit as might at first appear; as a break or loss of the suction column is extremely liable to produce serious damage to the engine by placing it in the position of exerting power enough at the steam end to do a certain amount of work and then being suddenly robbed of most of its resistance. Therefore it is best to allow, say, 5 ft. for a margin of safety, and so reduce the 13 ft. to 8 ft. This is again liable to reduction on account of the entering head necessary at the intake end of the suction pipe, and also on account of the bends in the suction pipe on its way from the well to the pump, and it is scarcely possible to get along with less than four changes in direction in the course of the inflowing water; one bend at leaving the well at the top of the vertical column; one in turning towards the pump near or within the building; one turning into the pump chamber; and one more in turning directly towards the suction valves. Of course some conditions and situations might reduce the number of changes in motion and direction to three, but not frequently, and in generalizing results it is always better to take an extreme view on the adverse side of the question when dealing with such an absolutely fixed factor as "suction lift," it being remembered that the hydraulic engineer

aims mostly to arrive at a safe conclusion so as to insure the results sought for, rather than to ascertain the exact balance of all of the factors involved. This latter accomplishment is absolutely impossible on account of the unknown changes in conditions and the inevitable errors in human judgment and workmanship, always present in practical operation.

The loss in head sustained by flowing water through a pipe is not great with properly designed bends, but is, nevertheless, something to be allowed for when dealing with such a small amount of fixed pressure; it depends upon the length or the shortness of the bend, although not in direct proportion, the resistance increasing rapidly in very short bends after the radius of the bend becomes less than one diameter of the pipe, culminating at a radius of bend equal to half a diameter of the pipe, which is the least radius possible to make in a pipe bend, as then the inner "corner" of the bend, so to speak, will be a perfect right angle in a 90 degree bend. After the radius of the bend reaches five times the diameter of the pipe the effect seems to be the same as though a straight pipe were used, but practically it will be good practice to allow one quarter or 0.25 of a foot head for each bend in discounting the atmospheric force for driving water into a pumping engine, as then it will be known that the flow of water will be without interruption from this item; with four bends this allowance will further reduce the net force for causing flow down to 7 ft. With the form of bottom to the suction pipe used by the writer there need be no allowance for entering head at the point where the water enters the pipe down in the well; this form being a bell mouth, with an outer diameter twice the internal diameter of the body of the pipe. If a foot valve is used, but which is unnecessary excepting at rare intervals and under some sort of abnormal conditions, of course there will be a resistance to be added for this item amounting perhaps to one foot, so that with a foot valve the net available force for sending the water to the pump on the basis of the length of a straight pipe equal to the total length of suction pipe, including length of bends,

would be 6 ft. under ordinary good conditions, with a total measured suction lift of 20 ft., and with all losses of head accounted for and deducted.

Regarding the resistance of water flowing around bends in pipes, there are well received conclusions based upon a formula from high authorities on the subject of hydraulics, but there is an amount of complex mathematics involved which would be out of place in a book of this character and aims, and, therefore, the writer in considering the mathematics of the matter, together with the factors of roughness and irregularity actually found in the work of making suction pipes for pumping engines, gives as a safe practical conclusion the allowance of 0.25 of a foot for each change in direction of the water in its progress from the pump well to the plunger barrel of the main pump.

Regarding the use of a foot valve, there is really no reason for such an appliance in a great majority of cases, as the writer has very fully demonstrated during the past three or four years in the installation of the largest and highest types of modern crank and fly wheel pumping engines. Even with the Worthington high duty engine, a machine which has always been supposed to greatly need a foot valve so as to insure the complete filling of the pipes and chambers with solid water before starting, there is at least one case wherein the foot valve was removed from the bottom of the suction pipe of a 6,000,000 gallon engine of this type, and under a 14-foot suction lift no inconvenience whatever followed. So that what was tried as an experiment was continued as a fixture for several years and continues so at present.

With 6 ft. as found above for a head to overcome the friction in the flowing water, well known formulæ will give a much higher velocity than is generally reckoned for water going into pumps and suction pipes, but the flowing of water into a pump is not continuous and uninterrupted as in the case of simply sending the water to a given point by a continuous flow. At the instant of the closing of the suction valves at the end of the stroke of the plunger, the water is completely stopped at

that end of the pump and must be promptly not to say suddenly started into the opposite end of the plunger chamber or into another plunger chamber which has just been discharged of its water. So that the velocity expressed in feet per second from a formula is largely discounted by the manner in which a pump actually takes its water, and the diameter of the suction pipe is brought up in practice to what has been observed as a fair practical figure, obtained by comparing the theoretical discharge with the ordinary actual rates of flow.

The final results of these determinations produce the following table of suction pipes for pumping engines of various capacities, based upon ordinary lengths of such pipes practicable to be used in regular work. If it seems to be necessary to use much longer suction pipes than are ordinarily found, say, considerably longer than 50 ft. for large pipes, it will be found the better practice to flow the water by gravity to a suction well nearer to the engine even at the cost of considerable trouble and expense. Of course longer suction pipes can be used when we are driven to it by the necessities of a situation, but the best results will not be obtained from the plant with extremely long suction pipes, even though enlarged diameters are employed to keep down the friction flow.

Table of suction pipes not over 50 feet, long and with not more than four changes of direction in the flow of the water from the well to the suction valves.

U. S. Gallons per 24 Hours.	Internal Diameter for 20 Feet Lift. *Inches.*
1,000,000	10
2,000,000	12
3,000,000	16
5,000,000	20
8,000,000	24
10,000,000	30
12,000,000	36
15,000,000	40
20,000,000	48
25,000,000	54
30,000,000	60
35,000,000	66
40,000,000	72

These sizes would answer for something more than 20 ft. suction lift, and would do no harm for less than 20 ft., so, perhaps, the table would answer as a standard table for regular work. The difference in cost would be trifling between these sizes and the least that could be gotten along with at all, and one case of trouble from too small a suction pipe would wipe out any saving possible in several cases, by reducing the diameter materially. And aside from this, the economy involved in the use of standard sizes in manufacturing is obvious.

The proper and complete bolting of suction pipe joints need hardly to be dwelt upon it would seem, but a brief mention of its importance will be in good place here, if for nothing more than to emphasize the great value of absolutely air tight joints where such a delicate effect as the maintaining of a perfect vacuum is concerned. Whenever it becomes evident that an air leakage does exist in some part of the suction connections, it is an almost endless task to discover and locate the trouble; it generally being found necessary to thoroughly paint the entire pipe with all of its joints, so as to cover the inlet of air wherever it may be, the difficulty of locating small but troublesome leaks being all but insurmountable. It is assumed by some that because the effects of pressure involved in the uses of a vacuum are very slight, the bolting of the suction joints need be only nominal, and not to be very seriously considered as compared with joints for pipes under pressure of the discharge portions of the pump; but in such a position there are many chances for annoying and expensive mistakes, and therefore the writer makes no distinction between the bolting of the suction and the discharge pipes so far as concerns number and size of bolts, the idea being that the loss of vacuum, although a rather delicate matter to deal with, is of too much importance to the well being and proper operation of the pumps to permit anything but the greatest means of assurance of complete and perfect workmanship.

Therefore the following table is presented as a guide in the laying out of suction pipe joints, and which, although costing

a little more than joints made with fewer and smaller bolts, will nevertheless pay to use where absolute assurance of air tight joints is desired, especially when it is considered that settlement strains, contractions, and expansions, and other possible distortions n the suction pipe, all tend to cause a letting up at the joints where the bolting is not solid and secure. This table is laid out upon a standard line for pipe joints, for pipes under heavy internal pressures such as force mains and the like, and will be found to give very satisfactory results in holding the packing in place between the flanges so as to prevent air leakages.

Table showing inside diameter of suction pipes; sizes of flanges; number and sizes of bolts; and diameter of bolt circles.

Diameter of Suction Pipe, Inside, Inches.	Outside Diameter of Flanges, Inches.	Finished Thickness of Flanges, Inches.	Diameter of Bolt Circle, Inches.	Number of Bolts.	Diameter of Bolts, Inches.
10	16	1	$14\frac{1}{4}$	12	$\frac{3}{4}$
12	19	$1\frac{1}{16}$	$16\frac{1}{2}$	12	$\frac{3}{4}$
16	$23\frac{1}{2}$	$1\frac{1}{8}$	$21\frac{1}{4}$	16	$\frac{7}{8}$
20	$27\frac{1}{2}$	$1\frac{3}{16}$	25	20	1
24	32	$1\frac{1}{4}$	$29\frac{1}{2}$	24	1
30	$38\frac{3}{4}$	$1\frac{1}{2}$	36	28	$1\frac{1}{8}$
36	$45\frac{3}{4}$	$1\frac{3}{4}$	$42\frac{3}{4}$	32	$1\frac{1}{8}$
40	50	$1\frac{7}{8}$	$46\frac{3}{4}$	36	$1\frac{1}{4}$
48	$59\frac{1}{2}$	2	56	44	$1\frac{3}{8}$
54	66	$2\frac{1}{8}$	$61\frac{1}{2}$	50	$1\frac{1}{2}$
60	74	$2\frac{1}{4}$	$68\frac{3}{4}$	56	$1\frac{5}{8}$
66	80	$2\frac{1}{2}$	$74\frac{3}{4}$	60	$1\frac{3}{4}$
72	88	$2\frac{3}{4}$	$82\frac{1}{4}$	66	2

CHAPTER XVIII

WATER PASSAGES AND PUMP VALVES

Where the suction pipe delivers the water to the suction chamber of the main pump, beneath the suction valves, the passages and openings should be ample and of favorable form to permit of the flow with the least disturbance and breaking up into small divisions until the water is actually delivered to the suction valve gratings or seats, where it separates into small streams for transmission through the seats beneath the valves, and into the plunger chamber of the pump.

Upon the return stroke of the plunger, the suction valves are closed, the water pressure is suddenly raised up to that in the force chamber or the space above the discharge valves, and so the water is forced out of the plunger chamber and out of the pump into the delivery pipe or force main.

The operation of pumping water is simple and well known and has been long known in general terms, but it is surprising how many bad and rough working pumps have been produced in this world. Many troubles have been met with by ignoring a few simple and necessary principles, among them being a desirability, not to say necessity, for direct lines for the passage of the water through the pump; the avoidance of reverse currents and places for lodgment of the air present more or less in all free out door water; design and construction required to easily and safely meet the sudden reversal from suction to discharge; keeping the water so far as possible moving in an upward direction continuously and uniformly; design and construction so that the working forces and pressures are properly met by corresponding support in the structure of

the pump body, or by transmission, safely and directly to the foundations or framing, etc.

The pump valves come in for a very careful consideration, and have been the cause of much debate and contention. Beginning in the far distant past, with one or a very few large valves, pump valves have nowadays been brought to the common practice by nearly all makers and in nearly all designs of pumps, of a nest or group of comparatively small valves, for the most part consisting of rubber discs varying in diameter, in thickness, and in hardness, but little for equal conditions or service. And by adopting these small valves great convenience and economy in manufacture of pumping engines are secured, principally because with a desirable size of valves, say from 3 to 5 inches in diameter, a greater or less number may be made to suit the demands for smaller or larger pumps, and at the same time permit the manufacture of valves and seats in large quantities, with the assurance that they can be used to advantage and economy. Previously to the advent of the Worthington pumping engine, about 50 years ago, the practice of very large single valves prevailed in water works pumps of considerable size and capacity, but Henry R. Worthington, then engaged in producing an engine to retain whatever advantages he could see in the Cornish engine then in its glory, and to omit the disadvantages clearly apparent, conceived the idea of replacing the single large valve with a number of small ones, an advantage in construction and operation quickly to be recognized and adopted by pump makers ever since.

Fig. 66 is an illustration of a typical pump valve for water works engines, and although this specific size and design is sometimes departed from, nevertheless the illustration indicates a valve which can be used in a very large number of pumps; and this being the case the great economy of its manufacture in special machines becomes apparent at once.

The valve seats are sometimes screwed into the valve decks, sometimes pressed in without the use of screw threads, sometimes bolted into place; but the manner of attachment most

favored, and no doubt justly so, for water works engine pumps, is the screwing in by a tapering thread of a brass valve seat into the cast iron valve deck formed within the pump body; the brass seat being sent firmly home into its socket by means of some sort of a power driven machine or tool.

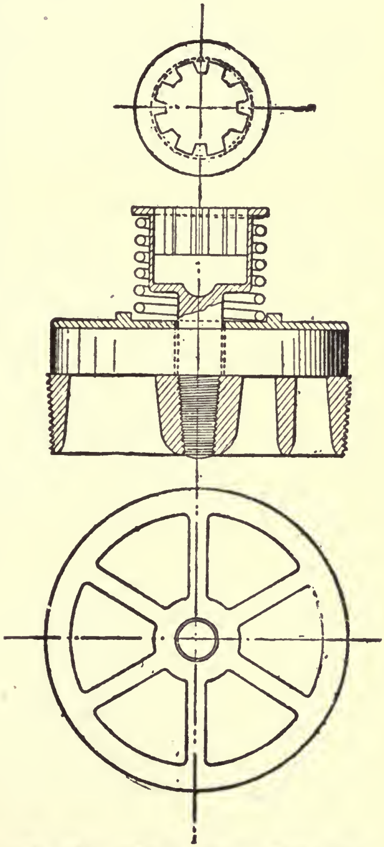


Fig. 66. — Typical Pump Valve for Water Works Engines.

The valve stems, which pass through the center of the valves, generally a separate piece from the seat, and in turn generally screwed into the center of the seat, is formed at its upper or outer end into a shoulder or guard to hold the spring in place and afford a hold for the wrench in putting in and taking out the stem when the valve itself is replaced.

Of course there have been a number of different kinds of pump valves used in the past, and some different ones are still used, but the general practice is pretty well settled down very nearly to the valve and seat shown in the illustration. For example, Fig. 67 is an illustration of another form of pump valve used a good deal at the present time. Its general features are the same as Fig. 66, as in this one also the seat, the central stem, the guard and spring, are of brass; but the stem and guard are separate, the stem being screwed into and then riveted at the under side of the seat; the guard holding the spring down

onto the brass valve plate is screwed onto the upper end of the stem and then a split pin or cotter is put through the stem above the guard to avoid all chances of the guard backing off; the edge of the plate on top of the valve is turned down around the upper outer edge of the rubber valve for support of the valve under pressure. In Fig. 67 the area is $4\frac{3}{4}$ square inches through the valve seat, and the diameter of the inner edge of the valve seat ring is 3 inches, making it

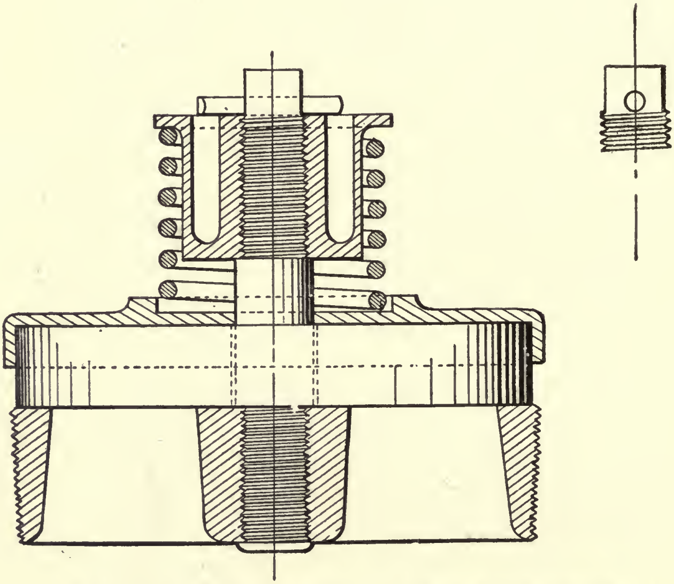


Fig. 67. — Typical Pump Valve for Water Works Engines.

necessary for the valve to lift one half an inch to give a rim area equal to the area through the seat; but the velocity of the water is no doubt greater at the rim than in the seat opening and so as a matter of fact in all probability a valve never needs to lift enough to make the rim area equal to the seat area.

The matter of total valve area, or total area of valve seat opening, has also been much talked about and disputed over but such area ought to depend upon the velocity needed to

pass the required quantity of water in a given time. Some authorities advocate a velocity not to exceed 3 ft. per second, and some set the limit at 4 ft. per second; but there are several factors to be considered. First as to the lift of the valves; the lower the pressure and the lower the rate of revolution of the engine, the higher the valve can lift; and to the contrary, the higher the pressure and the higher the rate of revolution of the engine, the less the valve may lift, if a smooth running engine is desired.

Assuming the valves to be not more than 4 inches nor less than $3\frac{1}{2}$ inches in diameter, the following table of extreme lift of valves and revolutions of the engine will give good results:

Water Pressure, Pounds.	Lift of Pump Valves, Fraction of an Inch.	Speed of Engine, Revolutions per Minute.
50	$\frac{1}{2}$	20
50	$\frac{3}{8}$	30
50	$\frac{1}{4}$	40
60	$\frac{3}{8}$	30
60	$\frac{1}{4}$	35
70	$\frac{3}{8}$	25
70	$\frac{1}{4}$	30
80	$\frac{3}{8}$	25
80	$\frac{1}{4}$	30
90	$\frac{3}{8}$	25
90	$\frac{1}{4}$	30
100	$\frac{3}{8}$	23
100	$\frac{1}{4}$	25
120	$\frac{3}{8}$	20
120	$\frac{1}{4}$	23
130	$\frac{1}{4}$	20
140	$\frac{1}{4}$	18
150	$\frac{1}{4}$	17

As is well known, many attempts have been made to have fast running pumping engines with high lift of pump valves, and in some cases fairly good results have been obtained, but seldom if ever do such engines operate without a great deal of noise and wear of pump valves; but contrary to this, when we see a quiet, smooth running pumping engine under fairly good water pressure, it will always be found that a very liberal area

of pump valves has been provided and consequently having small lift to pass the desired quantity of water.

As to the area through the valve seats or gratings, upon which the regular form of rubber pump valves rest, 3 ft. per second as the velocity of the water through the seats, for a general rule and in the absence of any special conditions, will be found to be very satisfactory and give a good, smooth running pump. A rate of 4 ft. per second will answer in many cases, and will at most times be acceptable where the pressure and the lift of the valves are not too high; but 3 ft. per second is to be preferred as the speed for smooth action and long life of the machine with low rate of repairs. The following table gives the net area through the valve seats for different kinds of pumping engines calculated at 3 ft. per second for the velocity of the water going through:

Area through suction valve seats for various capacities. Two double acting plungers.

Each Plunger.		Revolutions per Minute.	Travel, Feet per Minute.	U. S. Gallons per 24 Hours.	Water through Valve Seats, Moving 3 Feet per Second. Area of Suction Valve Seat Openings in Square Inches, for Each End of Each Plunger.
Diameter. *Inches.*	Stroke. *Inches.*				
9	18	45	135	1,280,000	48
12	24	36	144	2,400,000	90
15	30	31	155	4,000,000	153
18	36	27	162	6,000,000	228
21	42	24	168	8,600,000	323
24	48	22	176	11,800,000	443
27	54	21	189	16,000,000	601
30	60	20	200	20,000,000	784
33	66	19	209	26,500,000	998
36	72	18	216	32,600,000	1,220

For two single acting plungers the figures will be the same so far as diameter of plungers, revolutions per minute, and area of valve seat openings, are concerned, but the capacities per 24 hours will be only one half those given in the table, so

that double the required gallons per 24 hours will show in the table the dimensions for single acting plungers.

For three single acting plungers the figures will be the same so far as diameter of plunger, revolutions per minute, and area of valve seat openings are concerned, but the capacities per 24 hours will be only three quarters of those given in the table, so that one third added to the required gallons per 24 hours will show in the table the dimensions for three single acting plungers.

For example, if the plungers and suction valve area for 8,000,000 gallons per 24 hours is desired, and two single acting plungers are to be used, the 8,000,000 must be doubled, and then the diameter of plungers, revolutions per minute, feet travel, and valve seat openings found opposite 16,000,000 gallons. Or, if three single acting plungers are to be used, the 8,000,000 gallons is to be increasd one third or to the 11,800,000 as the nearest found in the table, and the dimensions opposite the latter figure will be correct for a three plunger engine for 8,000,000 gallons per 24 hours under the conditions given in the table.

With two double acting plungers in crank and fly wheel engines, the cranks should be set 90 degrees apart.

With two single acting plungers the cranks should be set 180 degrees apart.

With three single acting plungers the cranks should be set 120 degrees apart.

With the non-rotative or Worthington type of machinery, four complete strokes would be taken as one "revolution," although as a matter of fact there are no revolutions of anything, there being no wheel in its make up, but four strokes would rather be considered as one "cycle" corresponding to one revolution of the wheel engine. So that one "cycle" or four strokes, two double strokes at each side of the machine, is the equivalent of one revolution; it being noted in passing that there must be four operations in the non-rotative engine for each revolution, and two single acting or three single acting

plungers cannot be employed in this type, although four single acting plungers can be and are sometimes used. Therefore the non-rotative engine must be considered as a "four cornered" machine, or a machine with two double acting plungers and in no other class. From this it will be observed that the table as it stands accommodates the non-rotative engine, with the possible exception that this type is not often operated at quite so high a speed as that given in the table, but there is no perceptible reason why it could not be worked at the rates given with the proportions of diameter of plunger to stroke, and with the valve areas given. There is no hard and fast rule for plunger diameters, length of stroke, rates of revolution, and other details to be implied as absolutely necessary by this table, but the writer believes after many years experience, observation, and practice, that the dimensions and speeds given, will produce pumping machinery not to be excelled and probably not equaled as a paying investment where usefulness in proportion to the capital involved is concerned. All sorts of changes have been tried by various makers, but the record is held to-day, has always been held, and likely always will be held, for economy and long life, by proportionately large, slow running pumping engines. Repairs and stoppages are the most expensive items encountered in pumping water, and these should be reduced to a minimum by putting money enough into the machinery to enable it to handle its work with smoothness and ease. The Worthington old time, long stroke pumping engines have never been equaled for low repair accounts and absolute reliability for pumping water; they were run from 92 to 110 feet per minute and were generally built with a stroke of 4 ft. for 5,000,000 gallons capacity and upwards. The high duty engines of to-day of all types, which give the greatest economy, are long stroke, heavily proportioned machines. The only excuse for light weight, short stroke pumping engines is low first cost, the use of which involves a policy too often leading to disappointment and loss grossly out of proportion to the deceptive saving in capital at the beginning.

Having determined the proper area of seat openings for the suction valves of a pumping engine in proportion to the quantity of water, speed, etc., which will give the best all around results, least resistance and loss of quantity and power in passing the water through the suction valves into the pump, or more strictly speaking, into the plunger chamber, it will of course be certain that a similar application of areas to the discharge valves will give a like minimum of resistance and loss in getting the water out of the plunger chamber; so that the application of the same tables of valve details given for the suction valves will complete the pump so far as the inlet and outlet of water are concerned through the valves and valve seats.

As already pointed out, the passageways for the water within the pump body, from the suction pipe connection to the suction valves, within the plunger chamber, and beyond the discharge valves to the point of connection of the force main to the pump body, should be of liberal dimensions and of simple forms, so that the water may flow in as direct lines as possible and with the least resistance. It has been considered by the best and most experienced authorities to be a fair allowance to add one foot for loss of head for the suction valves, and one foot for the discharge valves, to the observed head noted by gauge and measurements, this allowance of two feet for the pump end of the machine to represent the resistance which is overcome by the power end of the machine in getting the water into and out of the plunger chamber, and which cannot be made to appear although the work is really done. The steam engine indicator, sometimes used for indicating the operations going on inside of a pump barrel, has a rather hard time at best, in handling such a stubborn element as water, and is wholly inadequate to demonstrate with sufficient exactness the actual state of the facts, beyond showing by a diagram the general conditions as regards either the presence or absence of abnormal and violent fluctuations arising from shocks in the water due to the sudden changes of form or speed of flow.

CHAPTER XIX

THE WATER PLUNGERS

In the natural course of things, the plunger is the next detail of the pump to be taken up and dealt with. There is not so very much to be said about the plunger; it is a cylindrical body sliding back and forth or up and down, either alternately into and partly out of the adjoining valve chambers as in a double acting pump, or into and partly out of a single plunger barrel or valve chamber as in a single acting pump.

The end or ends of the plunger moving within the body of the pump are shaped according to the ideas or fancies of the designer. Round nose, pointed nose, concavo-convex nose, and other forms of nose or end, are made for the extremity of the plunger which comes into direct contact with the water within the pump. The various shapes simply indicate the various ideas of designers in forming a plunger end which will offer the least resistance to the moving water, and move with the greatest smoothness and freedom from shocks, etc. It is not so far evident that any particular form of end has affected the efficiency of a pumping engine, and probably a half ball or half sphere is as good as any form for the end of a water plunger.

The office of the plunger is to attempt to form a vacuum by partly withdrawing from the valve or plunger chamber during the suction stroke of the pump, and then upon the return or discharge stroke, to form a pressure within the plunger chamber and the valve chamber above the suction valves, equal to or a little more than the pressure in the discharge or force chamber above the discharge valves; and then, having thus shut off communication with the suction chamber and opened up com-

munication with the force chamber, to discharge a quantity of water into the force chamber, equal in volume to the cross section and length of stroke of the plunger itself, or as nearly to such volume as may be, presumably amounting, in the best pumps and under the best pumping conditions, to 99½ per cent of the displacement volume of the plunger cross section multiplied by the movement during the discharge stroke.

The more or less perfect action of the valves and plungers control the hydraulic efficiency of the pumping engine, or in other words, determine how great a percentage of the calculated capacity of the pumps may be realized in the actual work of pumping water. In good practice, the loss which seems to be inevitable to some extent, in even the best machines, amounts to from 0.4 of one per cent to 1.5 per cent of the calculated displacement. Or putting it the other way, the efficiency or effectiveness of the best water works pumps varies from 99.6 per cent to 98.5 per cent according to the design and proportions of the pump, and the conditions under which the pumping is done. In specifying pumps, it is good practice to allow 5 per cent for slip, leakage, and other losses when designating capacity and sizes of plungers.

In the matter of the loss of displacement all types and classes of modern pumping engines are covered by two distinct and separate principles of operation, and under proper and reasonable conditions, and proper and reasonable proportions of machinery there is very little if any difference in efficiency in displacement. One of the principles of the operation of plungers is embodied in the performance of the Worthington non-rotative engine as originally brought out about fifty years ago; and this principle is that of the complete stoppage and pause of the plunger at the end of each stroke and so permitting the quiet seating of the suction valves and the complete closing of the communication between the suction and plunger chambers, before the return stroke of the plunger commences. The other principle is that found in all crank and fly wheel pumping engines, of the gradual slowing of the plunger as the crank

approaches the limit of its throw by virtue of the changing of the crank and connecting rod from an angle with each other to the same straight line, as the engine reaches what is known as the dead center. Summing up these two different principles as opposing each other, the former probably fills the pump the easier, while the latter probably picks up the load of discharge the easier; the total effect of both operations of suction and discharge working together so as to produce just about the same final effects regarding the losses in quantity as shown by actual delivery over a weir as compared with the amount of displacement called for by the cross section and length of stroke. A moments reflection will convince anyone that perfect displacement is impossible, and even impossible to calculate exactly; there is a certain amount of compressibility in the water as a practical fact, owing to the air which although ever so small in quantity is nevertheless present in all water handled by water works pumps; and the difference in the attenuating or expansive effect upon this air during suction, and the compressing effect upon this same air during the discharge, represents a change in volume which forever places out of reach of practical operations that complete measurement of discharge which is comprised within the expression of 100 per cent. It is also impossible to measure the diameter and stroke with absolute exactness. The writer knows by experience that the same man measuring three times, or three men measuring three times, or either one or three men measuring any number of times, will not always obtain the same record even with the finest kind of measuring instruments; and, of course, there is only one correct measurement, and an unlimited number of incorrect measurements, so that in actual operations it is found necessary to average several measurements and agree that this average is correct for purposes of settling questions under a contract. Even the gauging of plunger displacement by means of a weir is not free from uncertainties, as the reading, allowances, and calculations involved in the uses of a weir, with its crest readings, velocity of approach of

the water towards the crest, and other exquisite refinements necessary for correctness, and which often if not always have to be agreed upon from averages of several observers, represent disturbing factors standing between the facts and the nearest approach possible thereto. Taking it all in all the plunger displacement of a properly proportioned and adapted pumping engine is to be preferred to a weir demonstration, as being much

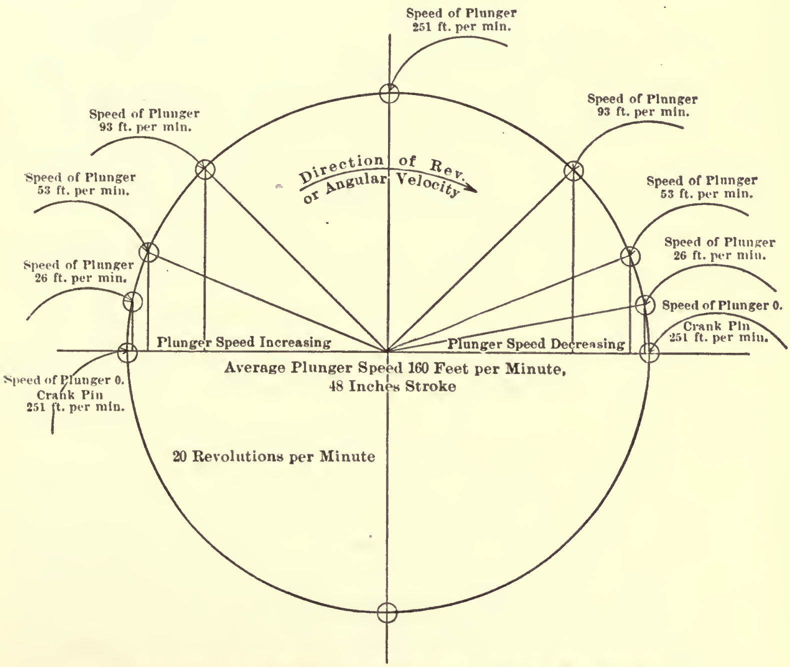


Fig. 68. — Diagram of Crank and Plunge Movements.

simpler and having its factors within easier reach of a more positive determination.

To illustrate the difference in plunger speed and movement between the rotative or crank and fly wheel, and the non-rotative or crankless engine, reference is made to Fig. 68, which shows the different positions of the crank pin of a crank and fly wheel pumping engine, and the corresponding positions of

the plunger at the same time, disregarding any modifications which would be made by the angle of the connecting rod. The engine is of 48 inches stroke and making 20 revolutions per minute, which would give the crank pin an angular velocity, or a speed around its circle of 251 feet per minute; found by multiplying 4, which is the stroke in feet, by 3.14, which is the ratio between the diameter and circumference of a circle, and multiplying this result by 20 revolutions per minute. The actual number of feet traveled by the plunger per minute is 160, found by multiplying 8, which is the number of feet in one revolution, or twice the stroke, by 20, which is the number of revolutions per minute.

Now with the crank pin half way between the two dead centers, the speed of the crank pin and the speed of the plunger would be the same; but, after the crank pin has reached half way between this middle point and the dead center, the crank pin will still have to travel 18.94 inches to reach the dead center, while the plunger will only have to travel 7 inches to reach the same point, which means that the crank pin will have to move at the regular rate of 251 ft. per minute while the plunger can reduce its speed to 93 ft. per minute, a reduction in the speed of the plunger of a little more than 62 per cent below what it had at midstroke.

When the crank pin has again cut the distance in half, between the 45 degree point, which has just been considered, and the dead center, bringing it within 22½ degrees of the dead center, its speed will still be 251 ft. per minute, but the speed of the plunger will be reduced to 53 ft. per minute, or a little more than 80 per cent lower speed than it had at the midstroke position.

When the crank pin has again cut the angle in half and arrived at a point 11¼ degrees from the dead center its speed will still be 251 ft. per minute, while the speed of the plunger will be reduced to 26 ft. per minute, a reduction in speed of more than 89 per cent, practically 90 per cent reduction from that at midstroke, and a plunger speed no doubt very much

below that of the non-rotative engine plunger at a corresponding point in the stroke.

Now turning to the non-rotative or direct acting pumping engine, as built in the early days when designed so as to reap all of the benefits and advantages of the system of pausing at the ends of the strokes, the speed is taken at 100 ft. per minute with a 4 foot stroke machine, making 12½ "revolutions" per minute. With a pause of one second at the end of the stroke, 25 seconds will be lost each minute, making it necessary to move the plungers during only 35 seconds of the time instead of 60 seconds contained in the minute. This will give an actual plunger speed of 175 ft. per minute, which is above the average speed of the crank engine plunger, and nearly 68 per cent of the highest speed of the crank pin as already observed. This speed no doubt continues just about up to the finish of the stroke or at least until the cushion of exhaust steam begins to stop the pistons; at all events the speed of the non-rotative plunger is likely somewhat faster near the terminal of the strokes than the plunger in the crank machine, and stops less gradually than the latter, but the pause is the saving factor in the operation.

With a pause of 1½ seconds, the actual speed would be 266 ft. per minute while the non-rotative plunger was really moving, as there would be only 22.5 seconds in which to do the work during each minute of time. And with a pause of only ½ second at the end of each stroke, the actual plunger speed would be 126 ft. per minute, as there would be 47½ seconds in each minute to get the motion.

With the non-rotative engine making 128 ft. per minute, which would be a 48 inch stroke engine making 16 "revolutions" per minute, and with a half second pause at the end of the stroke, the actual plunger speed would be 173 ft. per minute. With a ¾ second pause, the speed would be 213 ft. per minute.

With the non-rotative engine making 150 ft. per minute with 48 inches stroke, the "revolutions" would be 18.75 per minute,

and with a half second pause the plungers would travel at the rate of 218 ft. per minute; while the same engine making $\frac{3}{4}$ second pauses would have a plunger speed of 283 ft. actual travel per minute.

The speed of the plunger is generally given in terms of feet traveled per minute, and has long been, and is yet for that matter, a subject of much argument and dispute and misunderstanding. There is an extremely important difference in the way that plunger speed is produced, and this difference can be broadly shown by giving two ways of producing any certain or desired feet travel per minute. One is with a short stroke plunger, making comparatively a good many strokes per minute; and the other is with a long stroke plunger making comparatively few strokes per minute. The key to the situation is the rate of opening and closing of the pump valves per minute; and the speed which gives the number of times per minute allowed for the working of the pump valves, as determined by the best and most experienced makers, and as found in the pumping engines which give the highest efficiency and economy, is in the neighborhood of 20 revolutions per minute. That is to say, in crank and fly wheel pumping engines the rate of revolution of the fly wheels are as given above, and in the direct acting or non-rotative pumping engines, the corresponding motions of the pistons and plungers. The rates given in the table of suction valve areas will be found to be in line with the best practice and the best results obtained in actual work.

Upon the merits of the case, so far at least, everything is against rapid plunger action, that is, rapid rate of revolution of the engine and frequent reciprocation of the plungers. The longer the stroke of the plungers and the lower the number of strokes per minute, the better it will be for the effectiveness and durability of the pumps; and this principle can only be carried too far with steam machinery at least, with reference to the cost of the machine and for no other reason apparently. The records show that there is absolutely nothing in the line

of steam economy, in high speed in feet per minute or in rate of revolution. The steam engine holding the economy record to-day makes only 17½ revolutions per minute, and has a piston speed of only 197 ft. per minute. Its mechanical efficiency is 96 per cent and its steam consumption is 10.335 lbs. of steam per indicated horse power per hour.

Therefore the logic of plunger speed in a pumping engine is to make the engine as large and run it as slowly as the capital account will permit, or until the interest account and the repair account can be made to properly compromise with each other. The steam economy account may be safely left out of the question, or at least need not be feared in going towards large, slowly working pumping machinery.

The gas engine is showing great promise just now as to economy of coal, and is probably worth trying with a fast running pump if that is the only kind of pump practicable for it to operate, but the pump valves will doubtless need some modification from the present practice before the gas pumping engine can make a good name for durability; but if such a machine can show 200,000,000 ft. lbs. duty as easily as the steam machine can make, say, 120,000,000 duty, both with coal burned per 100 lbs., then it will pay to do some pump experimenting.

In water works pumping engines there are three well known forms of plunger application, the plunger and ring, the inside packed plunger, and the outside packed plunger.

The plunger and ring as the name implies consists of a plain, smooth cast iron plunger working through a brass ring, the internal surface of the ring where the plunger bears being generally grooved around its circumference, so as to provide a sort of water packing. The theory of the grooved ring being, apparently, that whatever water endeavors to leak past the plunger from the pressure to the suction end of the pump is compelled to pass first through the narrow space between the inner wall of the ring and the surface of the plunger; then when one of the circular grooves is reached, the suddenly increased

volume of the film of water retards the flow, and when the next division of close fitting space is met with the flow must again be accelerated; so that the alternate increase and decrease in the rate of passage of the film of water attempting to leak past the plunger results in such a complete retardation of the water that the plunger is reversed in its stroke before the leakage really has a fair chance of actually taking place. This form of plunger is mostly, if not always, confined to double acting pumps, or pumps taking in and discharging water at both ends alternately; the records and experience of pumping engine builders indicating that in fairly clear water, or water free from pronounced grit, such plungers and rings give very satisfactory results. In water containing fine, smooth, clayey silt, or a trace of vegetable slime, this type of plunger which with its ring is entirely without means of adjustment, will work for years with very little wear. The writer has seen and examined pumps with this form of plunger that have been working under favorable conditions during a period of five or six years, and has found the leakage to be very insignificant.

In a great many situations this type of plunger makes a very good pump, probably more of them being in use than any other form. It can be easily and accurately made, runs with little friction, requires no adjustment, and very little attention in fairly clear water free from grit in any considerable quantities. Its advocates argue that a plunger with a solid, non-adjustable ring, even if it does wear some, will really waste less water in the long run than a packed plunger, neglected and not given proper attention with reference to its being competently packed and operated; it being remembered that during actual operation in pumping water, the plunger is always moving in a direction opposite to that of the water trying to leak past it from the forcing to the suction end of the pump, which naturally enough helps to retard any water attempting to pass in a thin film between the surface of the plunger and that of the ring.

Fig. 69 shows the general features of this type of plunger with its ring and plunger rod.

An interesting experiment was tried by the writer some years ago with a steam fire engine, to find out how high a vac-

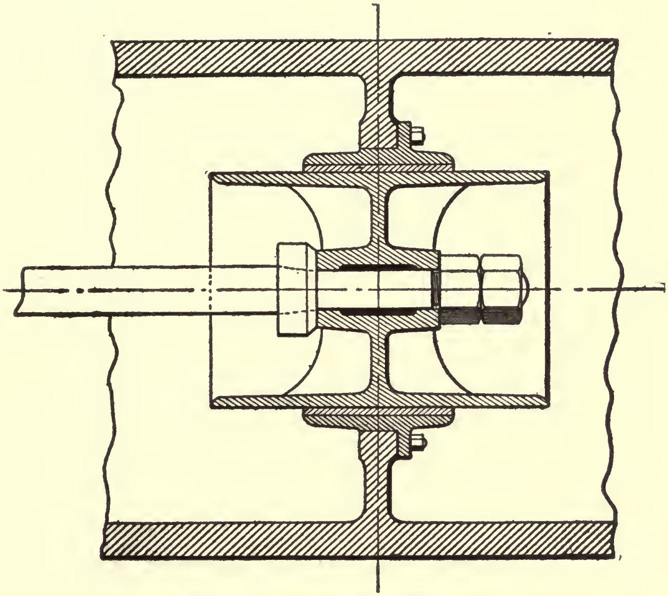


Fig. 69. — Plunger and Ring Pump.

uum could be formed and maintained with a solid plunger and ring. The plunger was of brass 5 inches in diameter and of 8 inches stroke. In this case the plunger was grooved and the ring was bored and then reamed as smooth as it was possible to make it. The plunger and the ring were both 8 inches long, and the ends of both were even with each other when the plunger was at its mid-stroke position, so that there were 4 inches of plunger in the ring at each end of the stroke. This test was made to learn how high a vacuum could be formed so as to judge of the capability of the engine to lift water 20 ft. when started with its pump and suction hose entirely empty, and it was the regular practice to make the plunger $\frac{1}{32}$ of an inch

free in the sleeve or ring, with reference to diameters. Such an experiment so far as known had never at that time been tried, and it was very important before entering a competitive trial that was at hand to know just what would happen. It was quickly demonstrated however that under perfectly dry conditions, with the exception of rubbing a very little oil on the plunger to prevent cutting before it got the water, a solid ring and plunger with a perfectly empty pump and suction, would form and maintain 21 inches of vacuum, equivalent to 23.1 ft. of water. As a result of the experiment, the trial was gone into, and with everything empty at the start, water was successfully and promptly lifted 23 ft., the suction hose of the engine strongly resembling the tall trunk of a tree in the eyes of the rather anxious builders of the machine and the skeptical spectators.

The inside packed plunger very strongly resembles the previously described plunger in water works engines, and also like the former is mostly used in double acting pumps of the horizontal class. In fact this construction is mostly the same as the last mentioned plunger, with the exception that instead of a solid ring for the plunger to work through, a complete stuffing box is substituted, fitted into the same position in the pump as the solid ring, but the back pump head must be removed for inspection and packing. It has long been a disputed question as to whether or not there is any advantage in having an inside packed plunger as against the solid ring variety; the argument being, that in the case of the ring the wear is very slow and its condition can be easily known and remembered, coupled with the certainty that it can only change slowly for the worse, while an inside stuffing box with its packing worn out, is a very leaky device, and it requires a good deal of close guessing and considerable attention to determine what the condition of the packing is, and to keep it in good order. In the writer's opinion the inside stuffing box is a very questionable device, and in fairly good water the solid ring is to be preferred.

Fig. 70 shows the form and construction of the inside packed plunger as generally employed in horizontal double acting pumps.

The outside packed plunger is considered by some authorities as the perfection of plungers, for the reason that leakages can

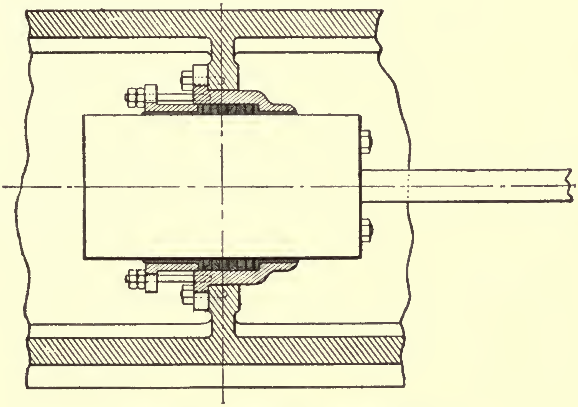


Fig. 70. — Inside Packed Plunger.

be seen and the packing is accessible for adjustment while the engine is running so as to stop such leakages. With proper attention this is so, and these very desirable results can be obtained, but as already pointed out, in good, clear water a solid ring which cannot be adjusted or neglected to its detriment, will leak the minimum amount of water for a long time; while in the very best water a neglected or badly adjusted outside packed plunger, even though right under the very nose of a careless attendant, may be allowed to leak a very serious percentage of the water pumped.

The outside packed plunger is used in nearly all forms of pumping engines, and in the horizontal machine is called a "center packed" plunger because, although outside packing is used, the packing of the plunger is accomplished by placing two stuffing boxes within the open space formed at the middle or "center" of the length of the horizontal pump body, one

stuffing box for each end of the pump. Sometimes the stuffing boxes are placed at the outer and inner ends of the horizontal pump body, with a solid partition at the middle of the length of the pump, to divide the end plunger chambers from each other. In this latter form the plungers are put in at the outer and inner ends of the pump, and connected together by means of heavy rods passing along the outside of the plunger barrels from end to end.

Nearly all of the vertical, crank and fly wheel pumping engines have single acting outside packed plungers driven directly from the steam cross heads by means of distance rods connecting these cross heads and the heads of the plungers rigidly together. There have also been made vertical engines

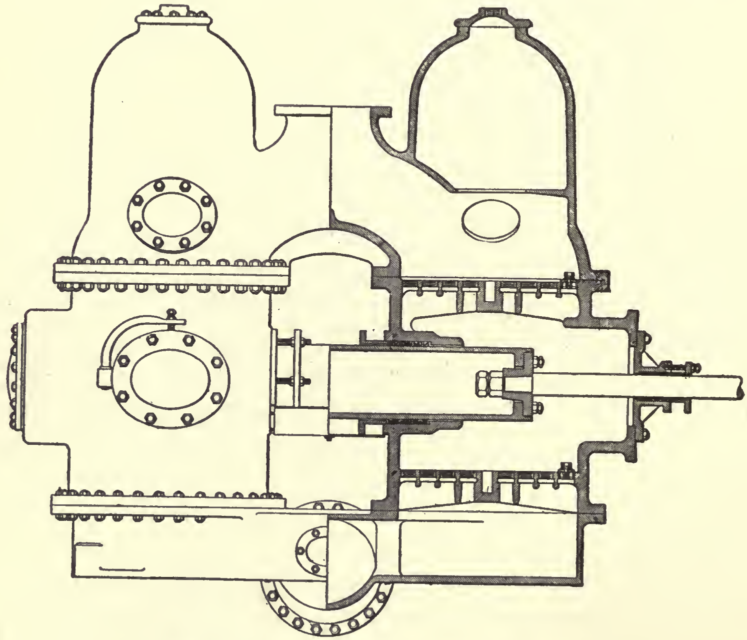


Fig. 71. — Center Packed Horizontal Plunger.

in which the outside packed plungers have been of the center packed class, and practically like the horizontal machine turned

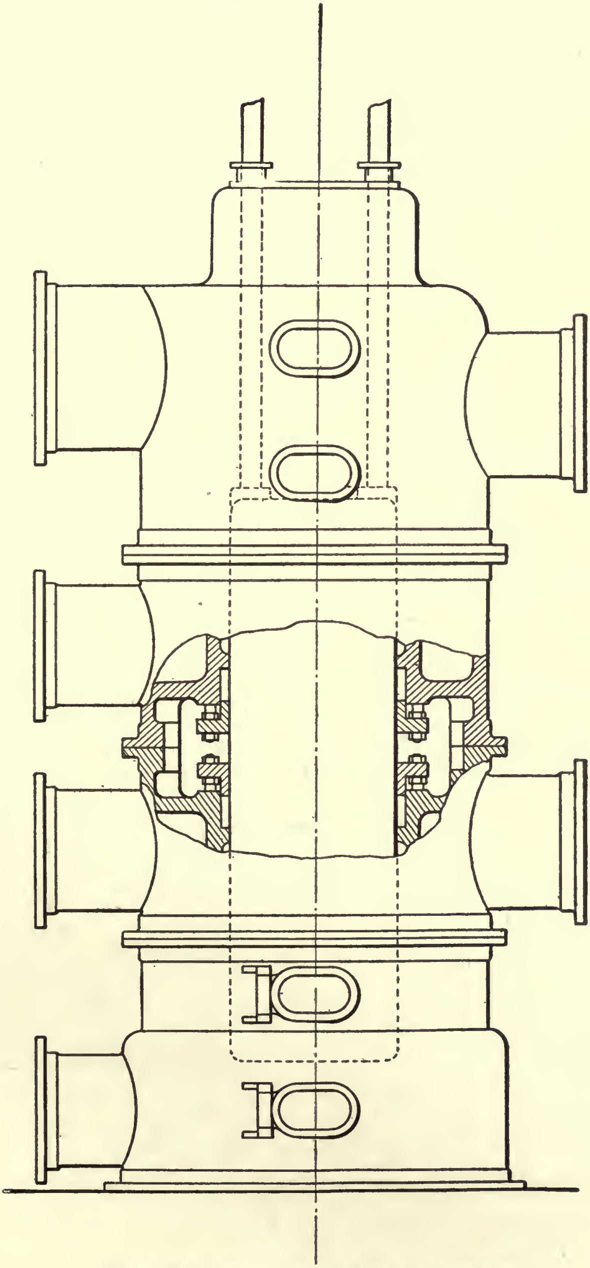


Fig. 72.—Center Packed Vertical Plunger.

up on end. Also, there have been built outside packed differential plunger vertical machines, with the upper half of the plungers smaller than the lower half, but with all plungers of the outside packed variety, these differential plungers usually being driven by means of a plunger rod passing through a stuffing box in the top of the upper plunger chamber.

Fig. 71 shows a center packed horizontal plunger. Fig. 72

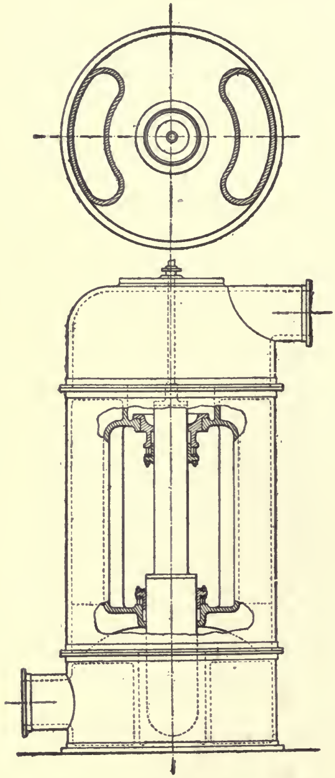


Fig. 73. — Vertical Outside Packed Differential Plunger.

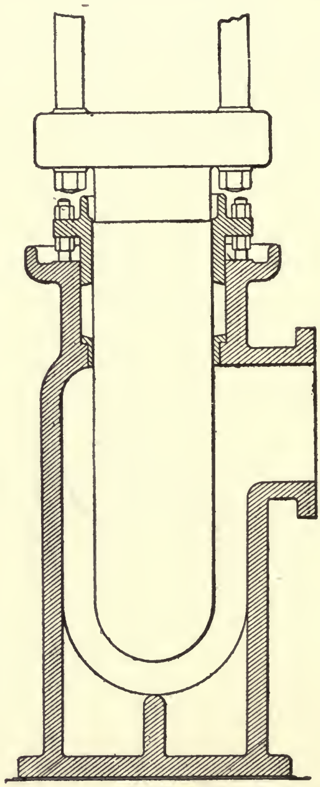


Fig. 74. — Vertical Outside Packed Single Acting Plunger.

a vertical plunger of the same class. Fig. 73 a vertical outside packed differential plunger. And Fig. 74 shows a vertical outside packed single acting plunger.

It goes without saying that the plunger is an extremely important part of a water works engine, and the maintaining of plungers and their packing in proper form and smoothness is well worthy of careful attention and some expense. They should be made of close grained material; most of them are made of cast iron, and as hard iron as can be worked with the tools in reasonable time. Late practice goes even so far as casting the plunger in a chill, and then after turning closely to shape and dimensions, grinding it to a finished surface where it works in and through the packing. When the value of low friction and high efficiency is considered, such refinements, really adding but little to the cost of the engine, are worth while obtaining.

The plungers working through solid rings in horizontal pumps are usually connected to the steam pistons of the machine by plunger rods extending from the plunger through a stuffing box in the pump head, towards the steam end, and connecting with the main cross head. The inside packed, and the center packed plungers are connected in horizontal pumps the same as in the ring type, by a rod through a stuffing box in the pump head, and so to the cross head. The outside packed, and the end plungers in horizontal pumps are connected together by side rods outside the pump barrel, and these rods are connected to the main cross head.

The outside packed vertical plungers, consisting of single acting plungers; double acting center packed plungers, and differential outside packed plungers, are usually connected to the steam cross heads by means of vertical distance rods extending from the plunger heads above the pump barrels, to the cross heads beneath the steam cylinders; these distance rods, or plunger rods, passing down through the space back of the cranks, between the cranks and the main pillow blocks, and also in front of the cranks. Sometimes two of these rods are used, placed diagonally from each other, and sometimes four of the rods are used, according to the ideas of the designer and the size of the machine. But whatever the kind of plunger

and however connected, the proper idea is to have a perfectly rigid and direct connection between the steam pistons and the water plungers, and so to avoid driving the load by or through moving connections and joints.

The plunger rods in horizontal pumps are secured to the plungers in various ways. Fitted into a taper socket in a hub at one end of the plunger and keyed; carried through the length of the plunger, with a shoulder or collar at one end and a large nut at the other, the nut secured after being solidly set home, by some sort of a keeper; by a nut, and shoulder or collar on the rod, the rod passing through a heavy partition or web at the middle of the length of the plunger.

In vertical pumps with double acting or differential plungers, and having plunger rods passing through the tops of the pump barrels, the hub and key at the top of the plunger are usually employed; while in vertical pumps having the plungers passing out through stuffing boxes at the top, the plunger head is usually formed so that collars and nuts at the lower ends of the distance or plunger rods secure the plungers. There are other ways of arranging plunger rods and connecting them to the plungers, more or less complicated and more or less dangerous in operation, but the foregoing methods cover the generally accepted practice to which the great majority of pumping engines have been reduced, and which are considered by different builders apparently as giving the best satisfaction.

CHAPTER XX

AIR CHAMBERS

The water is delivered into the force chamber above the discharge valves from the plunger barrel in absolutely separate quantities at each displacement of the plunger or plungers. And, in the absence of any equalizing device, there would be as many distinct shocks in the water column as there are strokes in the operation of the pump. But these distinct strokes of the plunger can be blended into a continuous and nearly uniform stream by the employment of a large vessel or chamber, placed in a suitable location above the discharge line of the pump, the upper portion of this vessel or air chamber being charged with or containing ordinary atmospheric air compressed to a tension like that of a spring, equal to the pressure of the water delivered by the main pumps. It is the great elasticity, or quality of instant compression and expansion, possessed by the air contained within the air chamber to suit the variations in pressure of the water being delivered by the plungers, which makes the air chamber such a valuable detail to a pump. There are many different arrangements of plungers, with reference to the different times and order in which they deliver the water through the discharge valves into the force chamber; and there is a wide difference between the regular or irregular action and effect in plungers in doing such work. But there is no water works pump working with plungers, delivering water into long lines of pipes or mains, however uniform, blended, and regular the actual plunger deliveries may be by virtue of the mechanical relation and arrangement of such plungers, but what the smoothness of the flow will be greatly improved by the use of a proper air chamber.

The extreme amount of irregularity would be produced by one single acting plunger taking in the water at one stroke and discharging it at the next; but the spasmodic work resulting from the use of such a plunger may be reduced to a very nearly uniform stream by a well formed and appropriately located air chamber near the delivery nozzle of the pump. One double acting plunger or two single acting plungers would naturally start the flow more steadily than one of the single acting form. Two double acting plungers with their strokes blended as in the non-rotative or in the 90 degrees crank and fly wheel engine, would be an improvement on one double acting. Four double acting plungers, or pistons, with the crank pins at 135 degrees and the center lines of the opposite sides of the engine at 90 degrees, as in the older type known as the Holly quadruplex, would give 8 impulses to the water column at each revolution of the wheel, and would give a still greater uniformity of delivering pressure than the 90 degrees crank engine. Three single acting plungers, with the crank pins set at angles of 120 degrees around the circle, will give a very good result in close uniformity of flow.

But with the best, and with all, of the different number and arrangements of plungers, the necessity for the air chamber exists where smooth work and high economy are desired in a regular water works pumping plant. The service which the air chamber performs is to receive the impulses of or pulsations given out by the plunger deliveries, against its air cushion, and thus take up by an elastic medium what would otherwise be a decided series of shocks within the mass of water and against the sides of the valve chambers and pipes. Or, in other words, without the air chamber the motion of the water would be intermittent; but as the velocity of the water on entering the air chamber is more than on leaving it, the level of the water therein rises above the outlet and compresses the air which fills the air chamber. Hence, whenever a plunger stops, the air thus compressed, reacting upon the water, forces it out during the momentary stoppage and thus keeps

a constant flow. The object sought in pumping water is to produce a steady uniform flow with the least change in pressure practicable; but the pump deliveries being from the nature of the machine intermittent and pulsating, on account of separate and measured quantities being repeatedly sent into the force chamber at different rates of motion, and at a rate of flow completely at variance with the uniformity sought in the force main, the water pressure in the absence of the air chamber would be suddenly increased for an instant at a time, and in effect a sudden and repeated increase in the velocity of flow in the force main at the pump would become necessary. The air chamber, however, when properly located, by means of its elastic cushion, safely receives the impact of the sudden rush of water from the pumps, and the attempted shock or rise of the water pressure is mostly expended in a slight compression of the confined body of air, and the surplus for the moment above what is required to keep up the constant flow in the main, is retained within the air chamber just for the instant necessary to permit of the uniform distribution.

In the practical construction of water works engines it is not always possible to locate the air chamber at the best point in the design itself, and convenience in the supporting of the various parts of the machine often takes precedence over any particular consideration of the matter of the air chamber being located so as to receive, in the best and most effective manner, the comparatively rapid deliveries and consequent pulsations of the plunger energy. It is very likely that the thought devoted to the air chamber is not so serious or abundant as it should be; but rather, the idea most prevalent is that after some of the other details have been arranged with reference to making a reliable, economical, and durable machine of moderate or low cost, some of the parts are made convenient for air chamber purposes; and this of course magnifies the importance of the plunger arrangement, on the principle that the less effective the air chamber, the more the plunger action must be blended to secure uniformity of flow.

This idea of undesirable plunger arrangement and insufficient air chamber combined, was clearly illustrated in the experience of the writer several times during the past few years, the opportunities for such observations not being any too plenty under actual water works conditions and service. In three pronounced cases the engine was of the two plunger class of the crank and fly wheel type; two single acting plungers, with the crank pins directly opposite each other, or at 180 degrees. One engine was on direct service with a closed system of pipes having no standpipe or reservoir. There had been some complaints about noises in the house faucets within the high service district supplied by this engine, and a call at a number of the houses resulted in the disclosure of the fact that every stroke and revolution of the engine could be counted very readily at the kitchen faucet, by the shocks or pulsations in the water, not taken up or obviated by appropriate air chamber capacity and location, although the engine itself apparently had a fair allowance of space above the discharge valves for entrapping air enough for a pump cushion.

In the next case, the plungers were also opposite each other in motion with the crank pins at 180 degrees, but this engine did its pumping into the distribution of the entire city; and the surplus over and above the consumption went into the reservoir, the city distribution being located between the pumping station and the reservoir. After the starting of the new engine, which is the one referred to, and which was the wrong kind of an engine for the service, many complaints began to come in from the water consumers; and there was no doubt whatever of the shocks and pulsations within the mains and service pipes. After the matter was carefully looked over and the conditions noted, a large air chamber was placed just inside the walls of the pumping station and connected with the force main outside of the check valve. The force main was 24 inches in diameter; the air chamber made of steel plate was 36 inches in diameter by 15 feet in height; the bottom of the air chamber being contracted to 12 inches where it was joined to the force main. The result

was most satisfactory. The pulsations gave no further trouble, and so far as the consumers were concerned ceased to exist, although there was no doubt still remaining, some traces of the rather positive action of two single acting plungers upon the water column.

The third was that of an engine pumping into a reservoir through a force main and with no connection whatever with the consumers. There were of course no complaints, but a person standing upon the ground at the reservoir directly over the force main where it entered the basin, could distinctly count the strokes of the plungers; the engine was about 2,000 feet away from the reservoir and was of the two plunger class with crank pins 180 degrees apart.

The writer is not aware that any remedy was ever applied in the first and third cases above mentioned, but there is scarcely room for doubting that ample air chamber capacity properly designed and located would have greatly reduced the pulsations in both cases.

In a water works plant the greatest use of the air chamber is really more on account of the distributing mains and pipes than of the pumping machinery, although the refinements of steam economy are found along the lines of smooth action, but no doubt the ordinary methods of construction furnish sufficient cushioning effects for the engine itself. Therefore the location of an air chamber outside of the engine construction should be made with reference to the force main where it leaves the building or approximately at that point; and where the air chamber is applied for this purpose, its dimensions should conform to the dimensions of the pipe with reference to its greatest probable capacity. There are no hard and fast rules concerning the size and capacity of such air chambers, but a proportion probably as good as any, and which experience shows at least so far will be sufficient to correct to an acceptable degree the pulsations of two single acting plungers, likely the worst condition to be met with, would be as follows:

The inside diameter of the force main where it leaves the

building is taken as the basis for the dimensions of the air chamber.

Then one and a half times (1.5) the diameter of the force main will be the inside diameter of the air chamber:

And seven and a half times (7.5) the diameter of the force main will be the height of the air chamber:

And half the diameter (0.5) of the force main will be the diameter of the opening into the bottom of the air chamber.

Such air chambers should be constructed along the lines of first class boiler work and made of steel plates; the bottom where it changes from its inlet opening to the full size of the air chamber could be made of cast iron for the smaller ones, and of cast steel for the larger ones; although if desired the bottom could be formed of the steel plate similar to the flanging of a boiler head, the top in all cases to be of steel plate like the top of a steam boiler dome.

With this rule an air chamber would be according to the following table:

Diameter of Force Main, Inches.	Diameter of the Air Chamber, Inches.	Height of the Air Chamber, Feet.	Opening in the Bottom of the Air Chamber, Inches.
16	24	10.0	8
20	30	12.5	10
24	36	15.0	12
30	45	18.8	15
36	54	22.5	18
48	72	30.0	24

The strength of such air chambers would have to be calculated according to steam boilers for thickness of plates, or as follows:

Multiply the pressure to be carried in the air chamber by the diameter in inches, and divide by 2, and this will give the bursting strain against the metal for one inch in length; then divide this result by one tenth of the tensile strength of the steel plate per square inch, and the result will give the proper thickness of plate to be used.

What thickness would be necessary in the steel plates for an air chamber 36 inches in diameter to hold a pressure of 100 lbs. per square inch?

$100 \times 36 = 3{,}600$, which divided by 2 gives 1,800 as the strain put on each inch of length of the air chamber, and what one inch will stand it will all stand. Good steel plate will stand 60,000 lbs. per square inch of section before parting, but for safety a margin of 90 per cent is allowed, reducing the actual strength reckoned upon to 6,000 lbs. per square inch. Then 1,800 divided by 6,000 will give three tenths (0.3) of an inch as the proper thickness of the plates for making the air chamber. This may seem like a liberal margin for safety, but the writer has seen water ram in mains send the pressure up from the normal pressure of 110 to the ram pressure of 350 lbs. per square inch, which would leave a margin for safety of only 2.85 to 1 with 60,000 steel and a thickness of 0.3 of an inch for an air chamber 36 inches in diameter.

The mention of water rams and shocks in force mains naturally brings forward the subject of check valves, which are generally placed in the line of the force main just outside of the main pumps of a water works engine. Check valves at times have their uses, but cannot be looked upon with favor under heavy pressure and with long force mains, where an extended "slug" of water is in motion away from the engine, for the reason that there is plenty of evidence that such mains have been caused to leak and even burst open by the reaction of the water column when the engine has been stopped a little too quickly, and the water column in reversing finds a suddenly closed check valve against it instead of the air chamber cushion which ought to be there for just such occasions.

In two plants in different places, recently enlarged, one with a force main 14 miles long, and the other about 7 miles long, there had been shown considerable distress in the matter of joints of the force main persistently leaking, with occasionally a break in the main. At different times and of course without any connection with each other beyond the facts of

similar conditions existing, these troubles took place from time to time, when the idea occurred to take out the internal details of the check valve at one of these plants, leaving nothing but the shell, and making of what remained of the check valve really a portion of the main. This gave the force main the benefit of the air chambers of the pumping machinery to react upon, and the results were most beneficial, in fact so much so that when a new and large additional pumping engine was installed, the check valve was omitted from the outfit; but there is a regular stop valve which can be closed when necessary and seems to answer all purposes.

With the omission of the foot valve also, this addition to the plant above referred to, seems to be flying in the face of all precedent, but in this as in other plants it has been fully demonstrated that under fairly good conditions easily obtainable in a great majority of cases, there is no need whatever of either foot or check valves being attached to pumping engines. Now both of the above mentioned plants have discarded both foot and check valves, and nothing short of coercion will bring them back again into use. It apparently requires some little courage to depart from such a long followed practice, but it would seem as though all of the well known characteristics of water under pressure favor the omission of check valves, which change certainly gives the force main the full benefit of the air chambers of the engines. If a check valve is used for any reason, there should be a liberal air chamber attached to the main outside of such a check valve.

CHAPTER XXI

STEAM PISTONS

It hardly seems necessary to call attention to the very great importance of the steam pistons in a pumping engine, or in any other sort of steam engine for that matter; but it is doubtful if Denis Papin had any idea of the importance of the steam piston and the figure it was destined to cut in the world when he conceived the idea so many years ago. The piston is simply a flat disc or plate with its edges bearing against the interior of the cylinder walls, and moving to and fro, backward and forward, or up and down as the case may be; impelled or driven by the excess of steam pressure brought to bear first against one side and then against the other side, by the alternate admission and exhaust of the steam by means of the valves and their mechanism.

There are two great things desired of a steam piston: To move with the least possible friction, and to keep the steam from leaking past. It requires very little pressure of the packing rings against the cylinder to prevent leakage, and it is a great mistake to endeavor to prevent leakage by setting the rings out tight against the cylinder walls. If a piston cannot be prevented from leaking by a very moderate pressure of the packing, or with very light springs to hold the packing out, then there is something radically wrong with the make-up or adjustment of the piston and its rings. A moment's consideration will show that there is no pressure or tendency under proper conditions of design and construction, that will tend to press the rings towards the center of the cylinder; but it will be shown that whatever pressure there is will tend to hold the rings in the position which the pressure hap-

pens to find them in when the steam enters the cylinder. Then if the cylinder has been bored smoothly and truly, all there is necessary for the springs to do is to maintain the packing rings against the cylinder walls with just as little force as will keep them there; and this required amount of force is very much less than is often produced by unnecessary and thoughtless setting out of the rings and springs.

The types and styles of pistons and their packings are many and various, — heavy rings, light rings, broad and narrow rings, continuous rings with a single joint, rings in segments or sections, rings with all kinds of cuts and laps, and with almost every kind and shape of spring. Pistons with one, two, three, and even up to five rings have been designed and made; and the working steam admitted behind the rings has been used to take the place of metal springs. In fact, it seems as though about all of the possible changes and combinations in this line that could be made to perform the office of a steam engine piston, have been tried. And although a few special cases have seemed to demand special arrangements and details, the piston which is used in the most, in the largest, and in the most economical pumping engines, is the plain cast iron piston with a single cast iron packing ring with one cut in its circumference. This ring is made nearly square in section, that is nearly as thick as it is broad, and is held out against the cylinder walls by a number of light steel springs without adjustment. The springs simply squeeze into place around in the annular space between the back of the packing ring and the bottom of the groove into which the ring is carefully although not very tightly fitted. The joint where the ring is cut to make it self-adjusting and elastic, is closed by what is known as a keeper, which is a sort of small flanged block generally made of brass, fitted back of the ring, so that the flanges of this block fit flush with the ring surface, into depressions cut into the surfaces of the ring which bear against the piston head and follower. This keeper or block forms a steam tight joint, and is held in place by one of the springs back of

the ring, or sometimes by one of the ends of two of the springs.

Perhaps as good a way as any to convey ideas upon the subject of steam pistons for water works pumping engines, is to describe several sizes of pistons for both horizontal and vertical engines; and therefore reference is made first to Fig. 75, representing a 16 inch piston for a horizontal cylinder.

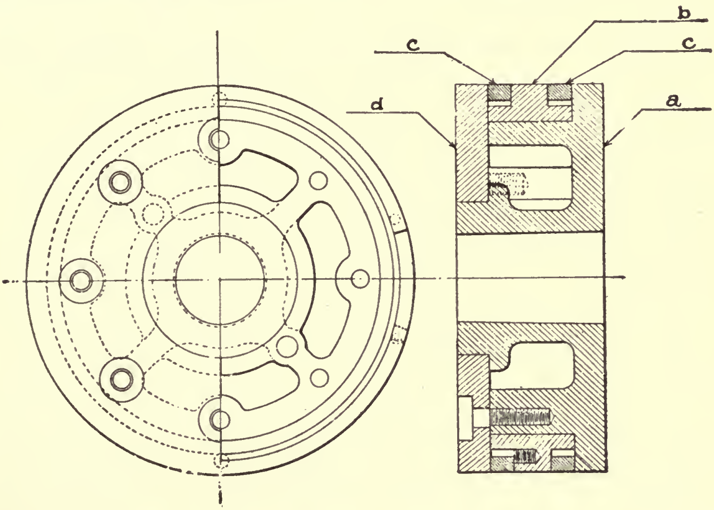


Fig. 75. — Steam Piston 16 inches in Diameter.

This piston consists of a main piston head *a*; a bull ring *b*, which accurately fits over a part of the piston head and helps to support the weight of the piston in the cylinder; two packing rings *cc*; and a follower *d*; all of cast iron. The piston head is snugly fitted to a tapered end on the machinery steel piston rod *e*, and is sometimes ground into place on the rod; it is held in place by a nut on the end of the rod. The bull ring and the packing rings are slipped into place and secured by the follower, which closes the end of the piston and is secured by 8 follower bolts $\frac{3}{4}$ inch diameter and $2\frac{5}{8}$ inches long under the head. The bull ring is **T**-shaped insection, and with the addi-

tion of the edges of the follower and piston head, gives ample wearing surface for carrying the weight, which is so distributed as to bring the pressure per square inch of bearing surface down to a very low figure, in this piston not more than 10 lbs. per square inch of actual bearing or riding surface. The packing rings are two in number, $\frac{5}{8}$ inch thick and with one inch face where they bear against the cylinder bore. This piston is of such moderate dimensions that springs back of the rings are not needed, the rings themselves being made so as to contain a certain amount of the spring effect by the following treatment:

The packing ring which is cast continuous, and with considerable stock to be taken off in finishing, is first turned to a diameter of $16\frac{9}{16}$ inches outside, and $15\frac{1}{4}$ inches inside. Then the ring is cut, and a piece $1\frac{9}{16}$ inches long is removed from the circumference. Then the ring is compressed until the ends are brought together, and is turned to fit the 16 inch cylinder; this will make the finished ring when released from the compression which it is placed under for the purpose of turning, $16\frac{1}{2}$ inches diameter outside, which when sprung into the cylinder bore, will maintain a very satisfactory bearing against the cylinder walls. The writer has used this self-springing ring in pistons as large as 25 inches diameter in condensing engines without the aid of any steel springs, or any springs aside from the tendency of the ring itself treated as above described; in rings as large as 25 inches, however, an additional ring inside of the wearing ring, made with the spring effect as well, will make a better piston.

There are two short steel pins tightly driven into holes which are drilled in the bull ring, for holding the packing rings in the same relative position to each other and to the bull ring, and preventing the cuts in the rings coming opposite each other. There are also two $\frac{5}{8}$ inch tapped holes in the bull ring for screwing in eye-bolts for taking out the ring; two of the follower bolt holes through the follower are tapped one inch, for screwing in draw bolts; and there are two $\frac{7}{8}$ inch tapped holes in the

piston head for screwing in draw or saddle bolts when it is desired to take out the piston.

Fig. 76 illustrates a 23 inch steam piston somewhat of the same type as the 16 inch, but in this one steel springs are used for holding out the packing rings against the cylinder walls. The piston consists of a piston head *f*, the follower *g*, the bull

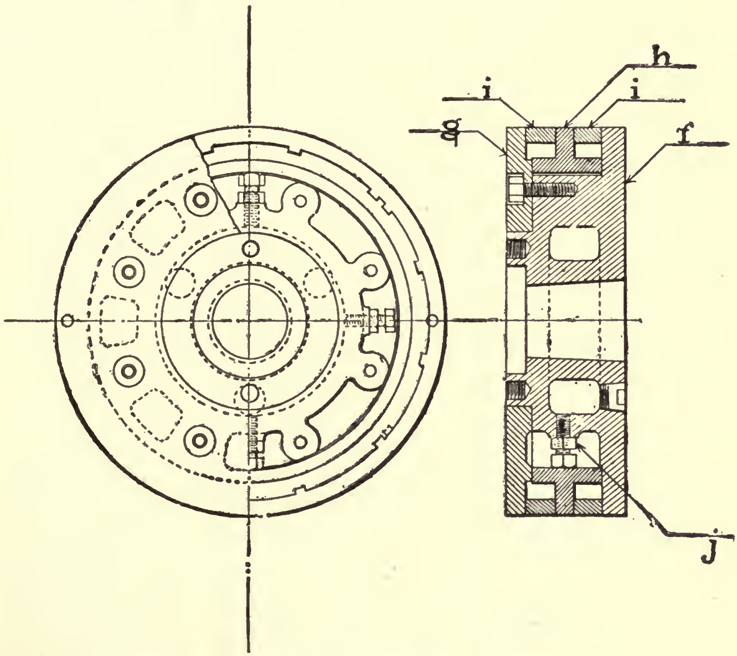


Fig. 76. — Steam Piston 23 inches in Diameter.

ring *h*, and the packing rings *ii*, all of cast iron. In this piston there are four centering or adjusting screws *j*, for adjusting the center of the piston into its proper place when from wear or any other cause it may be necessary or desirable. The piston head is secured on to the taper near the end of its rod by a nut in the usual manner; the bull ring is made with a loose fit on the piston head and is centered and secured in its position by the four adjusting screws already mentioned, these screws

being provided with lock nuts; the bull ring is of the same T-shaped section as in the 16 inch piston, and also helps with the piston and follower edges to carry the weight; the packing rings are separated by the edge of the bull ring, and immediately back of the packing rings and between them and the bull ring are located a series of twenty-two light steel springs for holding out the packing rings; the follower fits closely on the outer part of the piston head, holds the bull ring and packing rings in place, and is itself secured by eight follower bolts, $\frac{3}{4}$ inch diameter and $2\frac{3}{4}$ inches long under the heads.

There are twelve equally distant lugs formed on the inside of each packing ring for keeping the springs in place, eleven springs for each ring, one of the spaces being used for placing the cut in the ring, and for accommodating lugs and pins for keeping the rings in proper relation to each other; the packing rings are $\frac{7}{8}$ inch thick and $1\frac{3}{4}$ inches broad. There are the usual various sized holes tapped for eye-bolts, withdrawing bolts, and saddle for handling the piston, follower, and rings.

These two sizes illustrate a very good form of steam piston, especially for horizontal cylinders; and this general type is sometimes made without a bull ring, but with two packing rings rather thin in proportion to width, with their inner edges meeting together, and they are backed by one broad ring covering the entire backs of both packing rings; the springs are placed inside the broad ring and are usually supported on studs with lock nuts for adjusting the springs. This latter form makes a dangerous piston in incompetent hands, as they are likely under such circumstances to get set out hard enough to make a great deal of friction, and even score the surface of the cylinder.

This three ring class of piston with broad wearing or packing rings is sometimes made for horizontal cylinders, with solid supports for the lower third of the circle, and is provided with springs for the upper two thirds, the lower or solid portion being adjustable for centering the piston after wear has taken place, the springs sometimes made adjustable and sometimes not; but the general idea of the device is to provide a piston in

a horizontal cylinder which as it must ride on the cylinder surface in any event, needs no bottom adjustment beyond that due to wear, and whatever elasticity is needed better be provided in the upper sections of the circle. When in competent hands this form of piston is very satisfactory, requires but little attention, and wears well; it is easy on the cylinder, and altogether makes a very good piston.

Fig. 77 shows an entirely different type of piston. It is 30 inches in diameter, 12 inches deep, and this particular piston

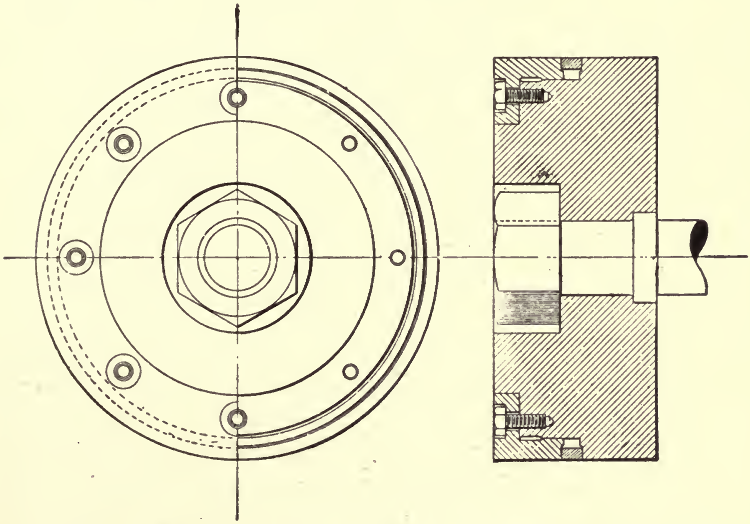


Fig. 77. — Steam Piston 30 inches in Diameter.

is for the high pressure cylinder of a vertical triple expansion pumping engine, but can be used for any other vertical cylinder of this diameter, as, for example, a vertical cross compound engine. It is made of cast iron, and in this case is solid so as to aid in giving the necessary weight to the moving parts of the engine with reference to balancing a portion of the pressure of the water column, usual in engines of this type.

The packing is of the single ring variety, made with one cut and provided with a keeper for filling the gap, as already ex-

plained. There is only the piston head, one packing ring, and a follower; the follower held in place by 8 tap bolts, $1\frac{1}{8}$ inches diameter and 3 inches long under the heads. The rod is made parallel and a tight fit in the piston head, the collar also fitting into a socket, and the rod secured by a large nut securely sent home. It will be noted that the corners of the rod at the junction with the collar are made with fillets so as to avoid possible tendencies to fracture at this point.

The follower is fitted with great care, with solidity and security in view; the upper part of the follower fits the top of the piston head, and is recessed at the upper corner of the head, and again fits for a short distance just above the packing ring; this makes accurate and solid work and at the same time makes the follower easy to get on and off the piston. The packing ring is placed within a space but a trifle deeper than its own thickness so as to keep it close to its work at all times, and prevent its being driven in by pressure when coming over the counter bore at the ends of the stroke, as has happened to packing rings at different times, causing mysterious noises difficult to locate; and by way of illustrating this point, the writer recalls a very curious experience in this line as follows:

A new large straight condensing engine had been built and installed by a concern which although doing the very best of work, had not had very much experience in large engines. This engine looked extremely well, worked smoothly so far as bearings and connections were concerned, produced beautiful indicator diagrams, and was altogether a very creditable turn-out for the shop that built it. But, just after turning each center, top and bottom, there was a loud thump or blow, which, although evidently not doing any particular damage so far as could be seen, was most distressing to hear, and most annoying to the builders and the buyer. Besides this, the steam economy was not as good as it should have been, and driving a flour mill, pretty close checks could be made upon the fuel consumption. About the only criticism that could be made, was, that at times, especially at short cut-off, the cut-

off corner of the diagram was rather too much rounded for best appearances and best results. A great deal of adjusting of the eccentrics, and changing of the valve rods and arms, were tried for several weeks, and in fact for a few months, but although now and then changing the sound, did not make any particular improvement. The top cylinder head was repeatedly taken off, but invariably everything was found in proper order. But finally it was noticed that the packing rings, two in number, had considerable space back of them, and depended entirely upon the springs to hold them out; and also that the counterbore was unusually deep and long, in fact so long that the rings overtraveled about $\frac{3}{4}$ of an inch. The idea was then suddenly evolved that the initial pressure drove back one of the packing rings, as it was so far beyond the real cylinder bore at the ends of the stroke, until the piston traveled a few inches, and then bang! would go the ring against the bore as it returned to its proper place. Some chocks or lugs were bolted into the bottom of the groove back of the packing rings so as to project between the springs and come within about $\frac{1}{32}$ of an inch of the back of the rings, and this completely cured the trouble both as to noise and waste of steam.

In Fig. 77 the springs are made as shown and put into place by a slight compression. The entire construction of this piston is massive, simple, and effective; the packing ring just free enough to adjust itself at all times against the cylinder, under control of the springs and with only enough lateral movement to insure its free action. The load put upon this piston by the incoming steam at the beginning of the stroke is 85,000 lbs. or about 43 net tons; and the weight of the piston complete with its rod is about 3,000 lbs.

Fig. 78 shows a 56 inch piston made in general features similar to the 30 inch just described, and differing only as would be natural in the larger form. The depth of this piston is 12 inches, the same as the 30 inch, and is the intermediate piston for the same triple expansion engines to which the 30 inch belongs, but would of course answer for any vertical cylinder

of 56 inches diameter. The follower in this case is only a ring with an outside diameter of 56 inches and an inside diameter of 44 inches, held in place by 12 tap bolts, $1\frac{1}{4}$ inches diameter and 3 inches long under the heads. The sectional view shows the piston head to be cast hollow, with seats at the center for the piston rod collar and nut, and with bearing surface at the outer upper edge for the follower; the right hand

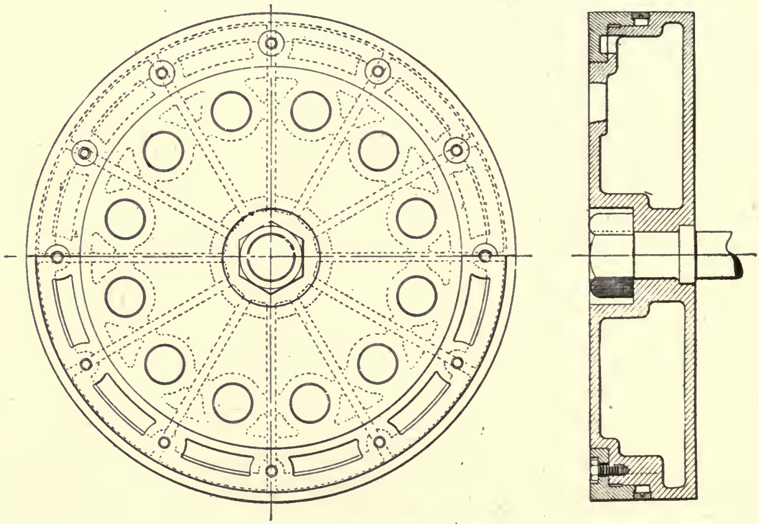


Fig. 78. — Steam Piston 56 inches in Diameter.

shows the bearing of the follower between the bolts, and the left hand the bearing at the bolt heads, and it may be noted how the follower is cut away at certain places so as to make it fit at the necessary points and at the same time be easy to remove and put in place.

The packing ring in this as in the last piston mentioned, is very limited in its movements, although perfectly free to adjust itself against the cylinder under the influence of the springs, which are located in the small space shown between the back of the ring and the bottom of the groove. There are 20 steel springs in this piston, quite moderate in their

pressure against the packing ring, in fact pressing only hard enough to keep the sliding contact perfect during the strokes of the piston. The plan view shows the ribs radiating from the central hub to the outer edge of the piston head, for strengthening and stiffening the construction; and also shows the positions of 12 plugs 4 inches in diameter where the openings were left in the head by the supports for the main core when the

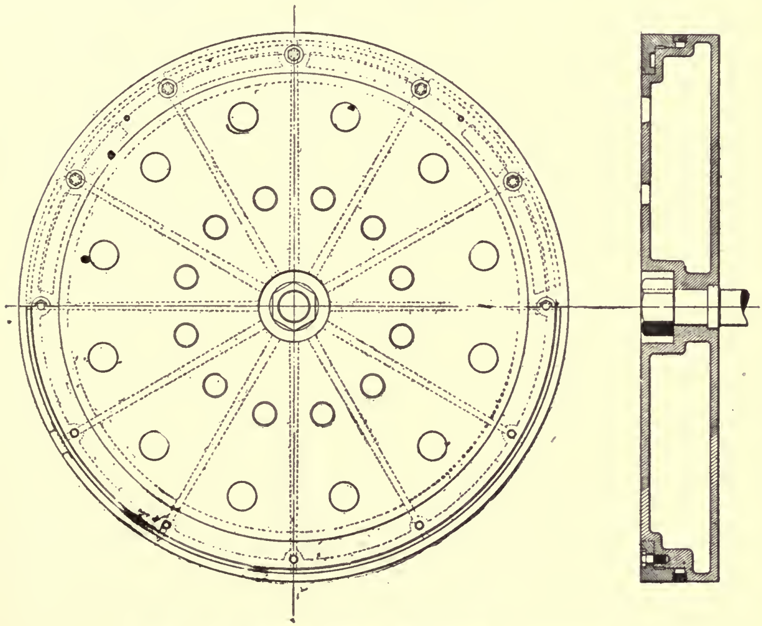


Fig. 79. — Steam Piston 84 inches in Diameter.

casting was made. The necessity for the great strength required in a steam piston may be understood when it is realized that the net load caused by the steam pressure at the beginning of the stroke, on this piston amounts to over 86,000 lbs. or about 43 net tons. This piston and its rod weighs nearly 5,000 lbs.

Fig. 79 shows an 84 inch piston closely similar in construction to the 56 inch piston just described, and has the same

depth of 12 inches; and in fact is the low pressure piston for the same triple engine the cylinders of which are 30 and 56 and 84 inches diameter, and of 60 inches stroke; but of course in a compound, or in a straight low pressure condensing engine, the same piston would answer.

The follower of this piston is also a ring with a diameter outside of 84 inches, and a diameter inside of 71½ inches. The follower bolts are tap bolts, 12 in number, 1¼ inches diameter, and 3 inches long under the heads. The section shows the construction and the plan view the location of the stiffening ribs, with two rings of core holes, one ring with three inch and one with four inch plugs. The construction, fitting, and bolting of the follower are the same as in the 56 inch piston, and also the general arrangement of the packing rings and springs; but in this large piston there are 30 of the steel springs for holding out the packing. The load upon this piston at the beginning of the stroke is about 53,000 lbs., and the piston and its rod weighs about 7,500 lbs.

These pistons are very perfectly made and fitted up, and even with the single packing ring give every evidence of the highest efficiency so far as concerns the two principal requirements, viz., working with the least possible friction, and keeping the steam from leaking past while the engine is at work. These particular pistons are in the cylinders of a vertical engine, so that with no weight bearing upon the interior surface of the cylinders, and with such slight springs as they have in them for holding out the packing, it can readily be imagined how easily and smoothly they must move in doing their work. Such pistons are sometimes made as large as 110 inches in diameter for low pressure cylinders of pumping engines, and from the high economy of such engines, (they hold the high duty record,) must be all that can be desired.

The foregoing illustrations, showing steam pistons from 16 to 84 inches diameter, really cover about all that would be necessary in producing pumping engines, small and large, horizontal and vertical; and these samples are taken from actual

engines with which the writer is familiar and has had to do in ordinary practice; such engines have been built and are running in regular daily service with perfect satisfaction. Of course, different designers hold various views about pistons; so they do about many other details; but it goes without saying that simplicity, strength, and effectiveness are very much to be desired in this class of work.

CHAPTER XXII

STEAM CYLINDERS

The proportions, design, construction, and arrangement of the steam cylinders of pumping engines have attracted and received a very large share of the attention devoted to the production of such machinery. And deservedly so. For the steam cylinder is really the heart of the engine, with the strokes of the piston representing the life beats of the system. The great study has been to get the steam into and out of the cylinder with the least amount of loss, and the greatest amount of work possible to derive from the heat supplied in the form of steam.

Strictly speaking, a proper cylinder is nothing more nor less than a perfectly round barrel true to the circular form and of the same diameter at all parts of its length. But there are numerous ways of construction, and of securing the cylinder in its place so that the piston may do its work in the most advantageous manner, and with the least practicable amount of friction in its movements.

Before the steam jacket came so generally into use, the construction of a steam cylinder was a reasonably simple part of the founder's and machinist's art. The steam jacket has been known a hundred years or more; but before the idea had been well grasped by steam engineers and steam engine designers that the heat was the vital factor, and not the mere brute force of the steam pressure displacing a piston, nearly if not quite all of the cylinders were made without jackets, the protection of the outside surface against outward radiation seeming to be all that was thought to be necessary. For very many years the steam pressures were low and the internal expansion of the steam within the cylinder was not carried out to any very

great extent. But with high pressures and high ratios of expansion, it can be plainly shown that the entire question is one of heat expended in proportion to work done, and that instead of this brute force pressure from the boiler to the piston, the steam is only the means by which the element we know as heat is carried from the place of combustion in the boiler furnace to the place of being converted into mechanical energy or work, within the engine cylinder. One of the clearest and most interesting demonstrations of the real cause of the power of steam as resulting from the use of heat, is given by a steam jacketed engine when its revolutions are increased or diminished by manipulating the jacket steam only, and independently of the steam going into the cylinder barrel where the piston is at work, the load, and all conditions of admission, steam pressure, and cut-off remaining the same. The greatest economy in proportion to the work done can only be obtained by a proper proportion between the amount of steam admitted into the cylinder bore, and the amount of steam admitted into the jacket space; but the fact that heat conducted or sent through the cast iron walls of the cylinder, from the steam in the jacket, will increase the mechanical work done by the engine, shows very clearly the real cause of the energy indicated by the engine. In other words, the question is in using steam jackets, how much of the steam shall we expend in the jackets, and how much shall we expend within the cylinder itself to obtain the best results?

This question naturally enough leads to a very serious and studious consideration of the arrangement of the steam jacket; its formation in practically making the cylinder, and the means of keeping the jacket space steam tight, under control, and supplying it with steam. There is little or no reason to doubt that in engines of fairly good size, say with 30 inch high pressure cylinders, and correspondingly large intermediate and low pressure cylinders if triple, and low pressure only if compound, the jacketing ot the cylinder heads is extremely important, and in fact more valuable than in jacketing at the sides of

the barrel, for reasons already explained in the chapter on steam jacketing. But for all that, the practice is generally to pay a great deal more attention to jacketing the sides and not the heads, probably for the reason that the sides are easier to jacket; and the question of the real economy of the steam jacket is not as a rule very thoroughly looked into. It is also very important to have steam jackets designed so as to obtain effective circulation of the steam or hot water as the case may be, and also to have arrangements so that the circulating may be absolutely controlled at will so as to adjust the work of the jackets to the best performance of the engine. Steam at boiler pressure is generally admitted into the jacket of the high pressure cylinder and the piping arranged accordingly; but it has become evident that the condensation from the high pressure jacket consisting partly of hot water and partly steam at a low pressure will give the best economy in the jackets following the high pressure; and it is also evident that when the steam is piped so as to go into the intermediate and low pressure jackets at anything like boiler pressure, there will be a loss instead of a gain in the total operation. As an example of an excellent plan of jacket pressures and connections, where, however, the cylinder heads were not jacketed aside from the steam which passed to and through the valves situated in the cylinder heads, the following may be noted:

Pressure in high pressure jacket, 151 lbs. main steam pipe.

Pressure in intermediate jacket, 40 lbs.

Pressure in low pressure jacket, 0 lbs. (Atmosphere.)

Jacket pipe 1¼ inches, from the main steam pipe to the top of the barrel of the high pressure cylinder, and out at the bottom of the jacket space. A steam trap at the bottom of the high pressure outlet pipe delivers the water of condensation to the pipe leading to the top of the low pressure barrel jacket.

A branch from the high pressure outlet, between the steam trap above referred to and the high pressure jacket, leads to the top of the coil in the first receiver, the pressure being regulated to suit economy; the outlet from the bottom of the first receiver

coil leads to the top of the intermediate jacket; then from the bottom of the intermediate jacket to the top of the low pressure jacket; the final outlet from the low pressure jacket leads to a water seal in the basement, the pressure being so low that no steam trap is needed on the low pressure outlet.

The drain from the body of the first receiver and a small portion of the working steam, enough to ensure all of the water leaving the receiver, is sent to the top of the coil of the second receiver; and from the bottom of the second receiver coil to the top of the low pressure jacket.

In coming to the matter of actual construction and operation, the factor of expansion of the iron of which the cylinder is made comes in for a great deal of attention, and it is pretty safe to say that in proportion to benefits derived, there is no detail of a steam engine which has given so much trouble, expense, and annoyance as the steam jacket. There are several ways of forming the narrow annular or circular space around the steam cylinder which constitutes the jacket space, but perhaps three methods will cover the greater portion of the actual practice in this line.

First: The casting of the jacket and the real steam cylinder in one piece, by making a steam cylinder to consist of two shells separated from each other by the jacket space, and held together and in proper relation by means of a series of ribs which form a part of the casting, and permanently connect the two shells together. In such a plan, openings at the ends of the cylinder, which are sometimes formed opposite the jacket space, necessary for the support of the cores which form the jacket space itself, are either plugged after the cylinder is finished, or left open, or a portion of them are left open, so that the side jackets, and the head jackets when such are used, may have free circulation of steam. Sometimes, where the cylinder heads are let several inches into the ends of the cylinder, so as to fill up the space between the steam ports and the cylinder ends, the barrel and head jackets are entirely separated by the position of the cores, and the circulation of steam is accom-

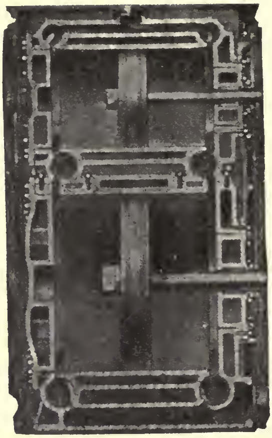


Fig. 80. — Section of Steam Jacket cast on the Cylinder.

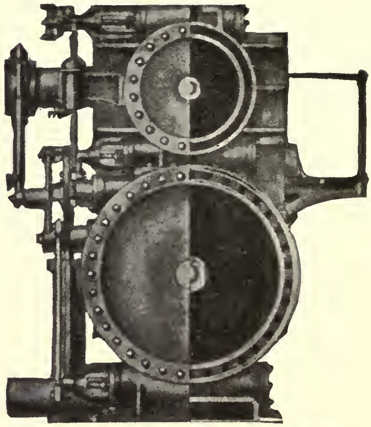


Fig. 81. — Section of Steam Jacket cast on the Cylinder.

plished by means of properly connected pipes outside of the cylinder and heads. Fig. 80 and Fig. 81 show sections lengthwise of the steam cylinder with a steam jacket cast on; and also a cross section showing the circular space between the inner and outer shells of the cylinder. In these figures both the sides of the barrel and the cylinder heads are steam jacketed.

Second: The making of the main shell of the cylinder containing the valve seats at each end, as a separate casting, and the inserting within this main cylinder casting an inner shell which forms the real cylinder in which the piston works. The inner cylinder only is bored, but there are bearings or fitting places at each end of the main casting for securing steam tight the ends of the two shells together, and thus forming the regular circular space for the steam jacket. (See Fig. 82.)

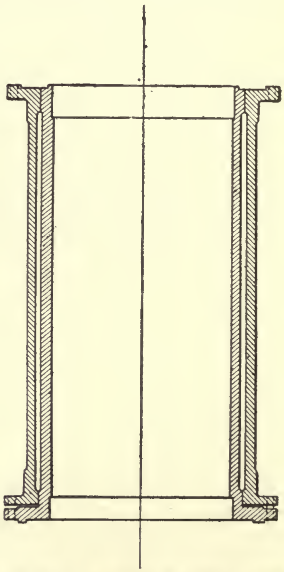


Fig. 82.—Reynolds Steam Jacket made Separate from the Cylinder.

The cylinder heads in this form of construction are some-

times jacketed and sometimes not; and the general arrangements of connecting the circulation of the jacket steam are similar to those for the jackets shown in Fig. 80 and Fig. 81. In the form of jacket construction shown in Fig. 82, the cylinder when planned for having its valves across the cylinder heads, as is found in important and large engines, is simply a plain flanged barrel of cast iron, fitted with an internal barrel for forming the circular space; both of these barrels finished

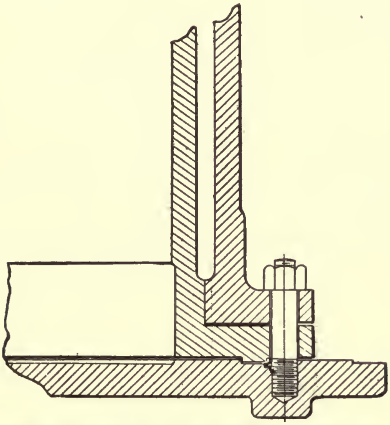


Fig. 83. — Bottom Cylinder Head with Reynolds Steam Jacket.

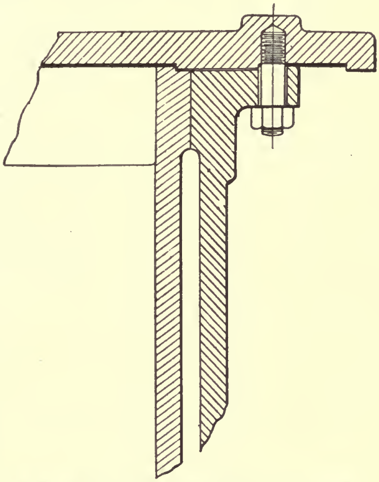


Fig. 84. — Enlarged Section of Upper End of Cylinder and Jacket.

flush at one end with a flange common to both for bolting on the cylinder head at that end, usually the free or outer or upper end of the cylinder. The end of the cylinder next to the engine frame, or what might be called the inner or lower end of the cylinder, is arranged slightly different from the other end; the main, or inner, or working cylinder, is formed with a flange which is arranged for bolting to the bottom cylinder head, and then the outer or jacket shell has a flange which is fitted on top of the flange of the working cylinder, one set of studs secured in the cylinder head passing through both flanges as shown in Fig. 83. Fig. 84 shows an enlarged section of the upper or outer end of the working and jacket shells.

In this form of construction, which is the most favored for both small and large engines of the better class, all of the steam and exhaust ports, valve chambers and seats, steam chests,

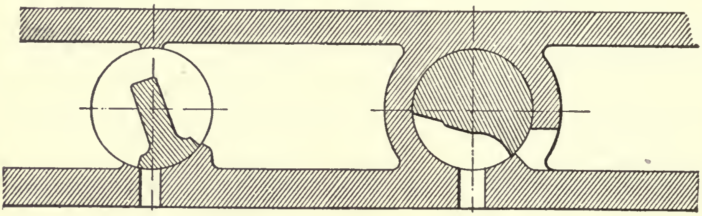


Fig. 85. — Section of Cylinder with Corliss Valves.

etc., are contained within the cylinder heads. (See Fig. 85 for sectional views of such cylinder heads, fitted with Corliss valves.)

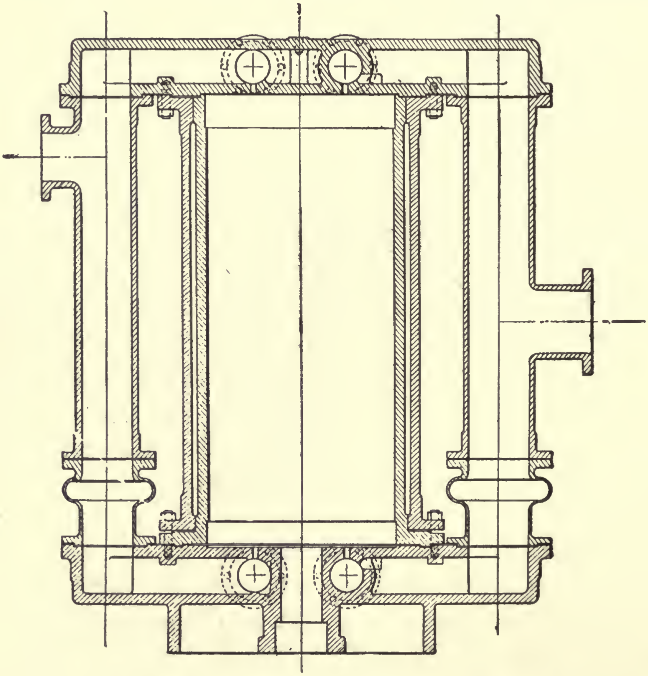


Fig. 86. — Section of Steam Cylinder with Side Pipes.

Fig. 86 shows a complete section of this type of steam cylin-

der with side pipes for steam and exhaust arranged with expansion joints, to compensate or allow for unequal expansion and contraction which is bound to be present in any construction employing several different pieces of metal fastened at their ends to the same terminal members, and where it is absolutely necessary to maintain steam tight joints.

In this form of cylinder the working and jacket shells are

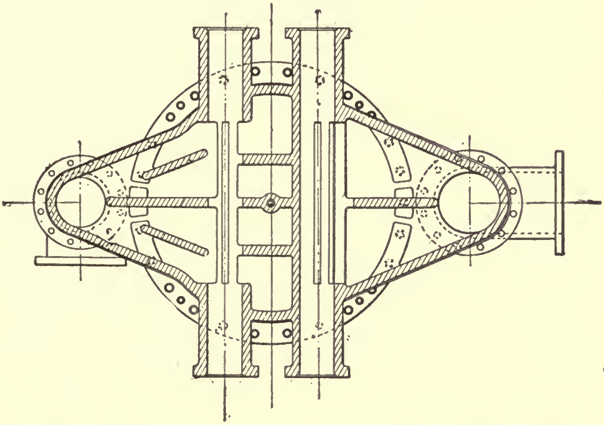


Fig. 87. — Cylinder Section with Ribs and Pipe Nozzles.

really formed into one complete barrel, expanding and contracting alike; but the steam and exhaust pipes are not only at different temperatures and pressures from the cylinders, but also different from each other. Hence, the rather complicated, but extremely efficient, scheme of construction.

Fig. 87 shows a section through the steam chest at the center line of the valve circle, and indicates the ribs for strengthening the flat surfaces, also showing the location of the steam and exhaust side pipes which distribute the incoming steam to both ends of the cylinder and take away the exhaust; the extreme outer flanged nozzle showing where the connections are made to the main steam pipe, a receiver steam pipe, or exhaust pipe, or a pipe to the condenser, according to the class and character of the cylinder or engine to which it belongs.

Where poppet valves are used for the low pressure cylinder of a triple expansion pumping engine, for the purpose of reducing the clearance or waste room at the cylinder ends to the lowest possible amount, the general construction of the cylinder and cylinder head is the same as already described, but of course the steam chest is very different from that of the Corliss type of valve. The idea of poppet valves is to have them close flush or even with the inside surface of the cylinder head so that when the steam and exhaust valves are closed, there will be no waste room on account of steam ports, as the

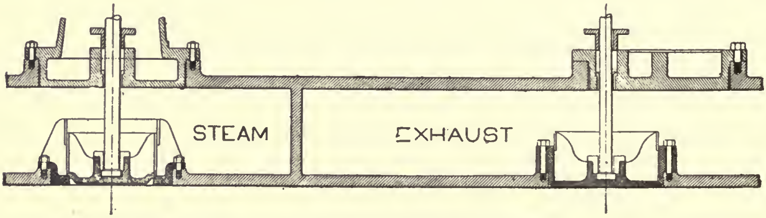


Fig. 88. — Poppet Valves Closed.

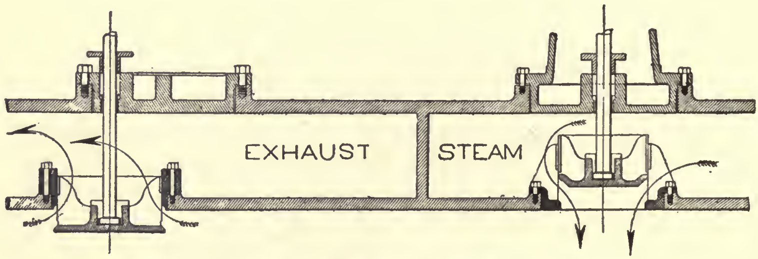


Fig. 89. — Steam and Exhaust Valves off the Seats.

only port is the circular opening left by the poppet valve when it leaves its seat. There is the clearance between the piston and the cylinder head, but this is brought down to a very small amount, and is in fact about all the loss of steam there is. The general idea may be understood by means of Fig. 88 and Fig. 89, the former showing in a simple way the position of the poppet valves when closed, and the latter showing a steam and exhaust valve off their seats; the steam valve opening

away from the interior of the cylinder, and the exhaust valve opening into the cylinder, it being remembered that the exhaust valve opens when the piston is at the opposite end of the cylinder from the particular valve opening, and will close before the piston reaches it on the return stroke; it will also be observed that even if the exhaust valve stem should stick in the packing or anything else hang the valve open, the pis-

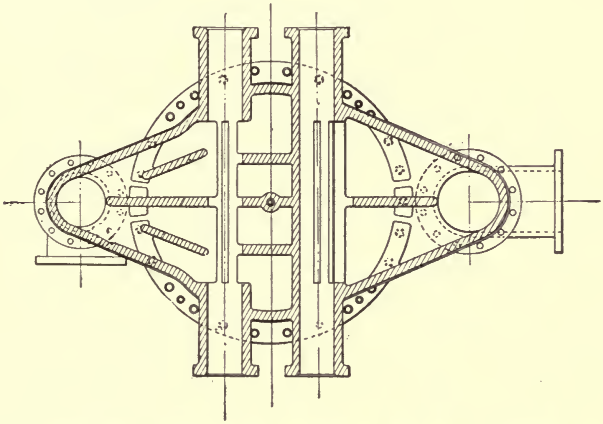


Fig. 90. — Ports of Corliss Valves Across the Heads.

ton will push it shut without doing any damage. The steam valve opens away from the cylinder and will never interfere with the piston.

With the Corliss type, even when the valves are placed across the cylinder head, the ports which carry the steam

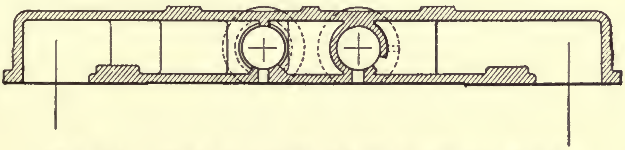


Fig. 91. — Ports of Corliss Valves Across the Heads.

through the head into and out of the cylinder represent waste space that must be filled with steam at each stroke of the piston. Fig. 90 and Fig. 91 illustrate these points, and the

ports show lost spaces which do not exist at all with the poppet form of valve. In high pressure or intermediate cylinders, this extremely fine point of clearness does not matter so much because the steam is used again in the low pressure cylinder; but the latter cylinder sends its steam to the condenser, and all waste space filled in this cylinder represents a certain loss which cannot be made good.

Twenty years ago, from 3 to 5 per cent of the cylinder volume did not apparently attract very much attention or adverse criticism as waste room, but in the light of experience the waste room has been gradually reduced by different arrangements of steam valves and ports until now the extreme refinement in this direction has brought the waste room down to $1\frac{1}{4}$ per cent in high pressure cylinders, and 0.8 per cent in low pressure cylinders in compound engines; and to $1\frac{1}{4}$ per cent in high pressure, 0.75 per cent in intermediate, and $0\frac{1}{4}$ per cent in low pressure cylinders of triple expansion engines. There is no special reason why the compounds are not calculated as closely as the triples, beyond the apparent idea that the compounds are not regarded as of so much importance as the triples, and are to a certain extent tolerated under special conditions where for some reasons it is not considered necessary or desirable to go to the refinements or cost of the triple machine.

The cylinder head joints and the piston rod packing has been gradually improved and brought to a great degree of perfection in these latter days of construction in the endeavor to reduce leakages and repairs to the lowest terms; and probably the desire to excel by builders of pumping engines for municipal water works, has been the greatest incentive to better and better work. Mill engines and electric railroad engines do not seem to offer the inducement to excel, probably because their performance cannot, in the nature of their load, be so minutely determined as in the case of pumping machinery, where the load can be brought to a constant quantity for any reasonable length of time for testing purposes.

The lower heads of vertical cylinders, and the inner heads of

horizontal cylinders, are strongly ribbed and made in special forms where they are bolted to the engine framing, this portion of the work of course being very important as the entire thrust and energy of the machine is alternately pushing and pulling upon the place of connection, constantly changing in intensity and degree as the pressure changes from initial to terminal. The clothing or covering of the steam cylinders to protect them from the loss of heat radiating from their outer surfaces, has long been recognized as a vital matter, and nowadays has been brought to a systematic and effective point. Probably one of the very best methods is to apply a plastic covering of some asbestos or magnesia mixture directly upon the heated iron, to a thickness of about $1\frac{1}{2}$ inches, and then outside of this place two thicknesses of $\frac{3}{4}$ inch hair felt. Outside of this comes the lagging or ornamental covering of steel or wood, partly for appearances and partly for the protection of the non-conducting material.

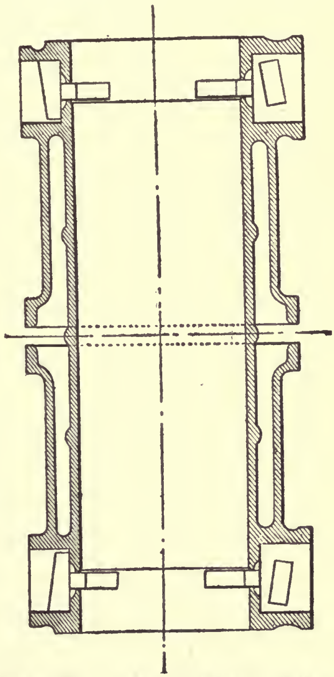


Fig. 92. — Leavitt Steam Jacket, Designed by E. D. Leavitt.

Third: The making of the inner or working cylinder in which the piston operates in one complete barrel, but with the outer shell which forms the steam jacket in two parts. These two parts of the outer shell are cast solid with the working cylinder at each end of the latter, but terminate a little short at the middle of the length of the cylinder so as to leave a gap in the jacket cylinder with the edges free. The opening so formed is closed steam tight by a copper expansion joint, the result being that as the inner or working cylinder expands and contracts by heat or the lack of heat, the

jacket cylinder can accommodate itself on account of the curved copper joint shown. (See Fig. 92 for the general section of the cylinder showing the working barrel and the jacket barrel; also see Fig. 93 for an enlarged section of a portion of the copper joint and ends of the jacket barrel.) This corrugated copper band forming the expansion element for the joint is secured in place with great care, by using two rows of comparatively large tap bolts at each side of the barrel opening; it is bolted

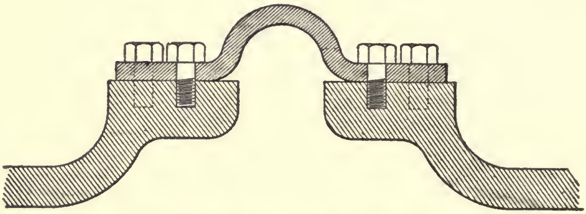


Fig. 93. — Enlarged Section of Leavitt Steam Jacket.

very firmly down, metal to metal, and caulked at the edges; the copper band being spun from a complete piece of metal and with no joint in itself. This expansion joint for a steam jacket has been found to give good satisfaction and is both scientifically and practically correct in its idea.

In this particular form of steam cylinder the valve or steam chests are located across each end at the side of the cylinder, and are arranged for flat gridiron slide valves, the valve seats being independent and bolted in place. Many attempts have been made and much money spent in machining, fitting, and scraping such valves to make them absolutely steam tight under working conditions, in some cases the work being finished with the cylinders hot so as to meet any distortions which may come from unequal expansion of the metal. Many cases have been observed in connection with certain details of steam machinery, in which the metal did not maintain the same form after being heated as it had when it was cold, especially where flat surfaces were being dealt with; such as these valves and their

seats have. The skill and quality of the work sometimes put upon these flat gridiron valves and seats no doubt represent the highest art of the machinist, and the valves are made perfectly tight under steam and apparently remain so for years.

But be this as it may, the record for the highest steam economy is not held by engines fitted with this flat sort of valve, but with a combination of Corliss and poppet valves, and without such refinements and expensive niceties of workmanship as is often devoted to the gridiron slide. The only reason possible to apply to such a state of things, is that with the absolutely steam tight slide valves there goes an abnormally large amount of clearance or waste room on account of the position in which it is necessary to place such valves so as to provide enough inlet and outlet for the steam. All engines must have some clearness between the piston and the cylinder head, but the poppet valve wipes out of existence all other forms of waste room, and no other form of steam valve does. The low pressure cylinder being the last cylinder, between the boiler and the condenser, under its very moderate pressure it is possible to operate large poppet valves and their mechanism; these valves also being quite easy to grind to a steam tight fit upon their seats. A pumping engine being so provided with such valves at the final outlet can afford to use valves in the other cylinders which perhaps may not be as perfectly steam tight as some other forms, although the Corliss valve properly proportioned and fitted comes pretty near, to say the least, to being a steam tight member, and coupled with this is the fact of its very low amount of waste room when placed across the cylinder head.

With this third form of jacket application in making the cylinder casting, any and all of the forms of valves, side pipes, and steam connections may be used, that are employed with either of the other forms mentioned above, although for an acceptable piece of work the second arrangement and construction would be the least costly. The matter of covering and lagging would vary only according to the construction

and arrangement of detail of the different forms of steam cylinders.

The quality of the cast iron and the accuracy of the boring, are factors in the construction of steam cylinders not easy to overestimate, and very likely do not receive the careful attention which their importance deserve. Cylinders should be made of strong close-grained tough iron, rather hard; in fact, if not brittle, just about as hard as they can be conveniently bored, and if they could be chilled and finished by grinding, so much the better, although if such work cannot be done and managed so as to be reasonably low in cost it might not pay. At all events, the actual work upon the cylinder bore is such a small percentage of the total work upon the engine, that it will pay to give it some special attention when the life and economy of the engine is considered, and when it is also considered what it means in the way of work and expense, stoppage, and annoyance, if it should become necessary to rebore the cylinders of an engine. In the construction of pumping engines, where high efficiency is looked for and where consequently the clearance of the pistons at the ends of the stroke must be closely figured, the boring of the steam cylinders becomes almost a fine art. The inside of the cylinder heads must be faced off to a true and smooth surface; the piston must be finished to an accurate thickness and accurately turned and faced so that the distance between the faces of the cylinder heads, will be equal to the length of the stroke of the engine plus the thickness of the piston and twice the desired clearance; and this clearance in quite large cylinders is sometimes brought down to $\frac{1}{8}$ of an inch at each end of the cylinder. The counterbore is a detail of cylinder boring not always appreciated, and to be correct should be carried to a point that will allow the packing to just nicely ride over the ends of the real bore, not more than $\frac{1}{8}$ of an inch, and closer than this even down to $\frac{1}{16}$ of an inch is sometimes figured and carried out in this portion of the work; in fact plans are sometimes drawn as fine as $\frac{1}{32}$ of an inch for

the overrun of the counterbore by the packing ring at each end of the cylinder, but it is very doubtful if such fine calculations can be carried out by even the most skilled workmen, when the making of joints and the changing of temperatures are considered. By an adjustment for length of the piston rod where it is secured in the cross head, the actual clearance between the piston and the cylinder heads may be brought down to a very fine point, and evenly divided, with temperature allowed for, by skillful men; but to have the overrun at the counterbores any certain amount and equally divided, at the same time, is calling for work almost if not quite outside of human accomplishment. The object of the counterbore is to allow the packing of the piston, that is, the ring or rings, to clear the ends of the real bore at each stroke and so avoid wearing shoulders in the cylinder surface, which would make themselves felt and heard when the length of the connections happened to be changed by taking up lost motion from time to time.

CHAPTER XXIII

CROSS HEADS

ONE of the details of a steam engine, and especially of a pumping engine, that is not very much noticed or apparently thought of, is the cross head. But, as it is the connecting point between the working parts of the steam and water ends, it is really a most important detail in the operation of the machine. In horizontal, low duty, direct acting water works pumping engines, that is, non-rotative engines of the Worthington type, the cross head acts only as a medium for connecting the ends of one, two, or three piston rods, as the case may be, to one plunger rod; one cross head and one set of rods at each side of the engine. In the regular machine as built on the original lines there was one high pressure piston rod, extending towards the water cylinder and entering the cross head where it was secured with a substantial key. Then, back of the high pressure piston the rod extended through a sleeve situated in a head between and common to the high and low pressure cylinders, into the low pressure cylinder and was secured to the low pressure piston.

The plunger rod was keyed into the cross head and extended toward and into the water cylinder where it was secured to the plunger by a nut or key according to the construction and size of the plunger. Later, the heads between the high and low pressure cylinders were made solid, and there were two low pressure piston rods for each low pressure cylinder, and, extending from the low pressure pistons through sleeves alongside of the high pressure cylinders, were carried towards the water ends and secured, generally by nuts, one for each rod, to opposite ends of forged wrought iron or steel cross heads extending

crosswise of the machine. In some makes of this type of pumping engine the cross heads were made of steel castings instead of the forgings, the principle of use being the same and the principal difference really being in the finish, as the forgings were generally finished and polished all over, while the steel castings were only finished at high parts. The plunger rods were connected the same as in the single cross heads already described.

In both of these forms of cross heads, the weight and that of the rods was carried upon the guide bars by means of a brass shoe attached to the under side of the cross head. The valve motion for the steam cylinders, and the air pump beams were driven from these cross heads, and the entire design was very substantial, neat, and effective. When the high duty Worthington pumping engine made its appearance and was reduced to a regular and commercial form, the general plan was retained, and the cross heads in addition to their other duties, carried the bearings for the compensating plunger heads, but the guides were very materially changed, and instead of a comparatively light girder casting located just below the center of the machine, the guides were formed within the massive cast iron frames which replaced the polished wrought iron distance bars, used in the older machines, as framing between the steam and water ends. Fig. 94 and Fig. 95 show these cross heads, of the original and the later forms of the non-rotative pumping engine.

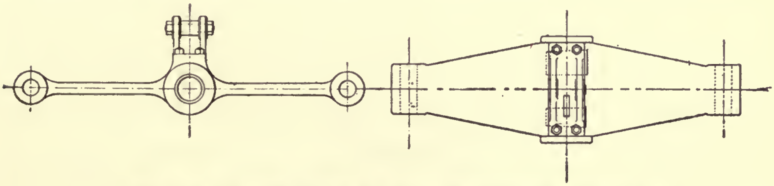


Fig. 94. — Cross Head of Early Worthington Engine.

In the first form of the Gaskill pumping engine, built by the Holly Manufacturing Company, the cross heads were of the locomotive type, sliding upon square bars, these bars also

assisting in stiffening and holding the machine together. The sliding portion holding the adjustable shoes was located above the jaw containing the cross head pin for the high pressure

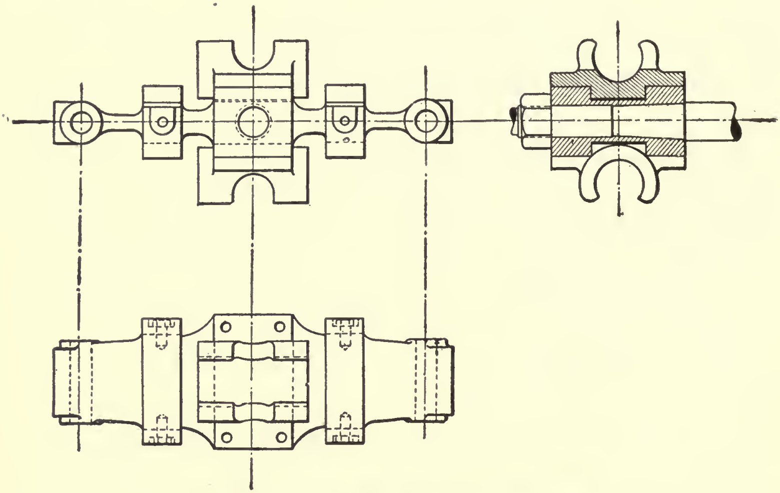


Fig. 95. – Later Worthington Cross Heads.

link journal; and below the wrist pin for the low pressure cross head; thus leaving the ends of the connecting rods and links free, clear, and very accessible. In the later engines of this type the open framework, somewhat similar to that of its non-rotative rival, which marks the earlier machines, gave way to cast iron girder frames in which the guides were formed; and the cross heads then took on a different form from the originals to conform to the changes in the framing. In these later cross heads, sliding blocks in each frame supported forged cross pins which were simply prolongations of the regular cross head pins. With the later cross heads, the main connecting rods were lengthened so that instead of working upon pins at the tops of the vertical rocking beams, they worked upon the pins secured in the cross head, above described. This change gave a longer connecting rod without lengthening the engine, but changed the motion so that the compression in the steam

cylinder at the closing of the exhaust valve, and at the moment the engine passed the centers, was less effective for smooth running.

Fig. 96 shows the early and Fig. 97 the later cross head of this type of machine. In the older type the high pressure cross head was keyed to the piston rod, and provided a wrist pin

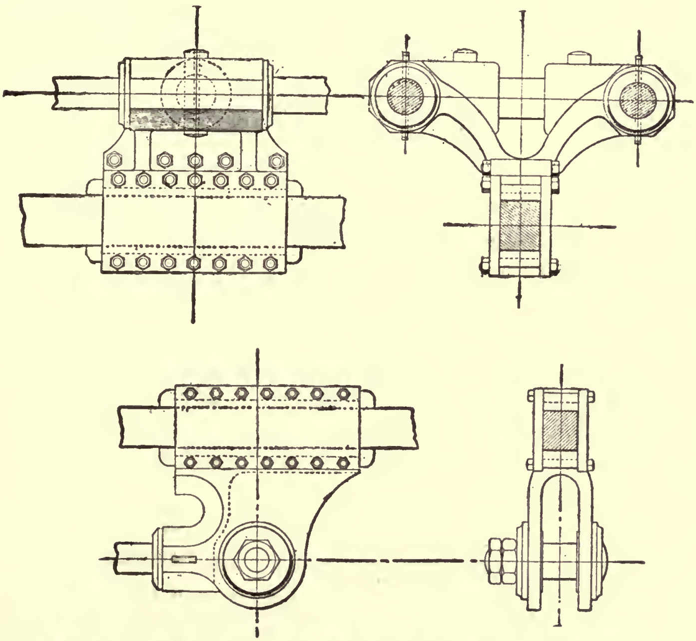


Fig. 96.—Early Gaskill Cross Heads, High and Low Pressure.

for the link which carried the power to the top end of the rocking beam, and so the high pressure cylinder had no direct drive upon the water plunger, but its power was partly transmitted to the fly wheel through the connecting rod which coupled to the top end of the beam also, by a forked end; and partly to the lower link which carried the power from the rocking beam to the plunger rod. The low pressure cross head was formed so as to receive two piston rods from the low pressure cylinder, and between the hubs into which these rods were secured,

there was located the wrist pin for connecting the lower beam link already mentioned; the drive from the low pressure piston

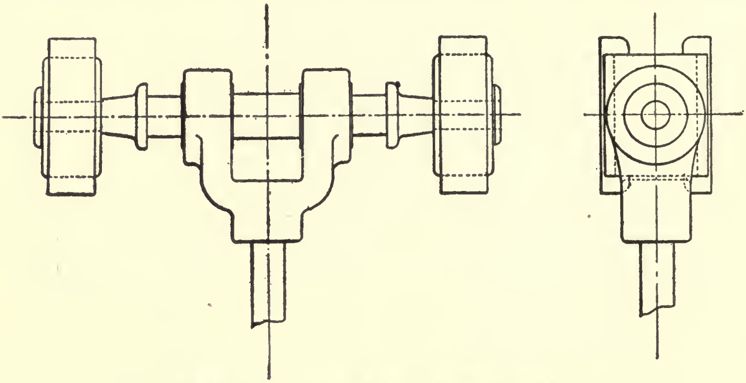


Fig. 97.—Later Gaskill Cross Heads.

was directly to the plunger as the plunger rod was keyed into this cross head.

In the later engines, the high pressure piston was keyed to the cross head as before, but the guide bar just above the cross head was omitted, and the guides were formed in the cast iron side frames which took the place of the round bars; a jaw was formed in the cross head which took the end of the connecting rod instead of the link, and the link connection to the rocking beam was made by two links one at each side of the cross head connecting with journals at the outside of the beam; this did away with the fork ended connecting rod, but made two links necessary.

In the horizontal, cross compound pumping engines, which have now attained the dignity of a pronounced type, as brought out by the Allis-Chalmers, the Platt Iron Works, and Snow Companies, the cross heads are quite different from those already described. In these cross compounds, the cross heads are formed with jaws for taking the connecting rod, and travel on guides in the lower part of the frame in the first and last mentioned engines, liberal hubs being provided for securing distance rods which extend past the crank shaft and

into corresponding hubs in the plunger cross head, these distance rods being placed diagonally with reference to the cross section of the engine, one rod above the shaft and back of the crank, and the other rod in front of the crank below the center line, thus forming a rigid drive from the steam pistons

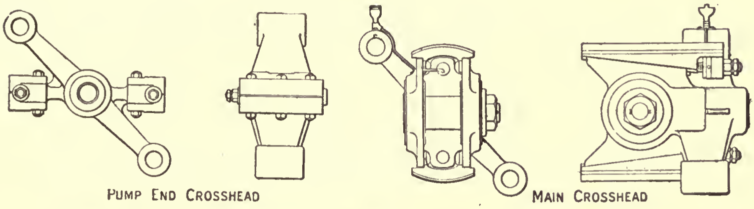


Fig. 98.—Cross Heads of Cross Compound Pumping Engines.

to the water plungers directly through the cross head. This makes one of the most direct acting crank engines possible to devise, as the surplus work of the initial steam is sent straight into the fly wheel, so to speak, and when the wheel gives back the work in equalizing the expansion of the steam, the connecting rod sends it straight to the cross head and so into the plunger.

Fig. 98 shows the general construction of the cross head of the cross compound horizontal pumping engine.

In vertical pumping engines, the conditions and applications of the power to the work of pumping are considerably changed from those of the horizontal machine; although the idea of transmitting so far as possible the work of the steam pistons directly to the water plungers is carried out rather better in the vertical machines. The Worthington type is for the most part, the horizontal machine stood upon end and the moving parts balanced against gravity by means of air pressure within what is called the balancing cylinders; there being no cranks and connecting rods attached to the cross head, of course the advantage cannot be taken of balancing one side of the mechanism against the other, as would be possible with a vertical

cross compound engine having cranks opposite each other, or with a triple expansion having cranks 120 degrees apart.

The Worthington vertical engine has been built mostly of the high duty class, and the cross heads besides forming the usual medium for joining the steam pistons and the water plunger rods, carry the bearings for the journals of the compensator plungers; the cross heads being guided at each end

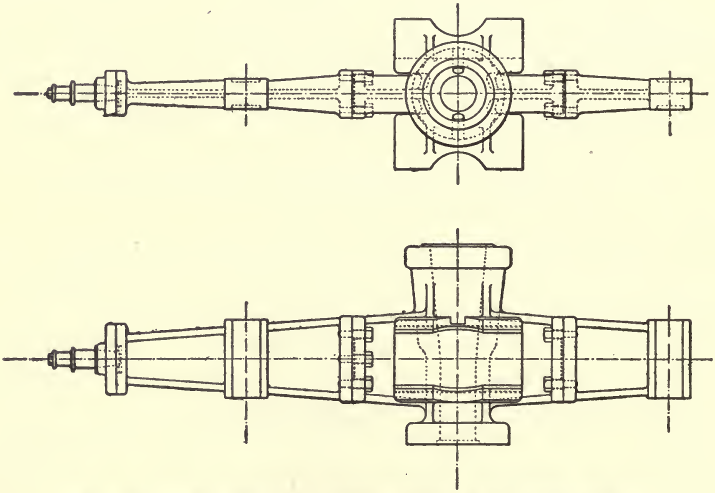


Fig. 99. — Worthington Vertical High Duty Cross Head.

by special guide castings secured to the framing below the bottom heads of the low pressure cylinders; the main framing being entirely independent of the guides. From the outer ends of each cross head there are driven the various side rods which give motion to the steam valve gear, each cross head operating the main valve motion for the opposite set of cylinders, and the cut-off mechanism for its own set of cylinders. This cross head is shown in Fig. 99.

In the vertical, triple expansion, and cross compound, crank and fly wheel pumping engines, at first, fifteen or twenty years ago, forged iron and steel cross heads were used, but of late years steel castings have been introduced, and as the latter

are the less costly to fit up, with the present day facilities for getting good annealed steel castings, they will probably hold the preference in the absence of some special reasons or desire for the forgings.

Although the more costly, the forged cross heads for the vertical engines have many attractions in design, accessibility, and appearance. The center of the connecting rod bearing or wrist pin and the trunnions for receiving the guide shoes, come so naturally and conveniently in line that they can be turned on the same centers, thus insuring absolute sameness

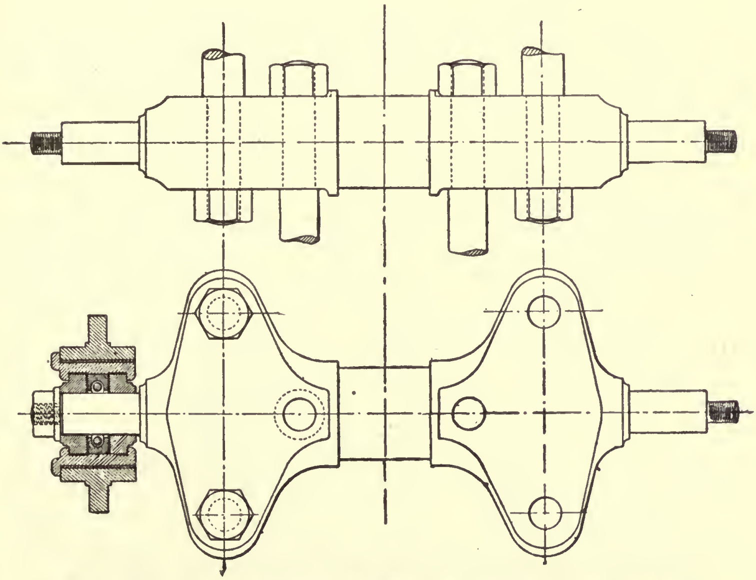


Fig. 100.—Forged Steel Cross Head for Vertical Triple Engine.

of center lines so desirable in cross heads. Also, the natural design for a forging lends itself completely to the accommodation of the distance rods extending from the cross head to the plunger head below; also giving a very convenient place for attaching either two piston rods, or a bridge tree for a single piston. And withal we have the comfortable consciousness that the cross head is a forging from the steam hammer with

all of its parts and material thoroughly condensed, known, and in sight during construction and operation. The forged cross head had its advent with the double "A" frame in vertical pumping engines; but since the introduction of the single "A" frame, and the tower form of frame, the guides are so formed and placed that the original type of forging does not fit into the construction so well. And, as the single frame, and the single piston rod which so naturally accompanies the single frame, can be made more economically than the double design, the ever present evolution which brings the lowest cost for an equally reliable result is bound to hold sway. There is no room for doubting the superiority of the single "A" frame, for its strength, its appearance, and its economical production in the foundry and shop.

With the single "A" frame and its internal guides there

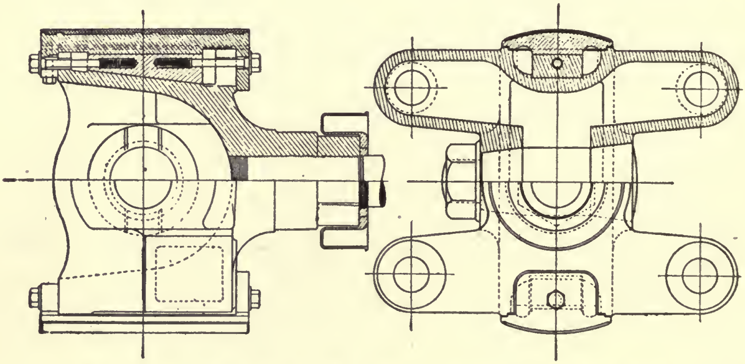


Fig. 101.—Cast Steel Cross Head for Vertical Triple Engine.

comes rationally enough an entirely different form of cross head, a form difficult and very costly to make of a forging, and so the builder must turn to something that can be made in a mold, and that means an annealed steel casting, as cast iron cannot escape both riskiness and clumsiness at the same time.

Fig. 100 shows the steel forged cross head now passing away, with all of its beauty, finish, and mechanical grace. Fig. 101

shows the cast steel cross head with its intense practicability both as to low cost and adaptation to the more economical construction, although it cannot entirely hide its tendency towards a clumsy appearance. But handsome is as handsome does, so long as the design is accurately proportioned and the steel casting properly annealed.

Cross head shoes are made mostly of brass, of cast iron, and of cast steel; with various means provided for adjusting for wear on the guides, mostly of the wedge type, controlled by set screws and binding bolts. The most general and likely the best practice, is to have cast iron guides and babbitt faced shoes, with the surfaces so proportioned as to bring the pressure resulting from the angular position of the connecting rod at each stroke below 25 lbs. per square inch of the wearing surface of the shoe.

There are other forms of cross heads for pumping engines, than those herein shown, more or less special for special situations; but these are the ones generally used, and as they give excellent satisfaction and are at least as low in cost of production and operation as can likely be made for a good article, they will probably continue in practice in this line of work for a long time to come.

The form of the cross section of the guide at the surface where the shoe slides, is made in various forms: the "V" shape; circular like the interior of a cylinder; and flat. These are the principal forms, with the circular probably predominating in the latest practice, or during the past fifteen years or so.

CHAPTER XXIV

FRAMES AND BEDPLATES

To make a general statement of the case in hand: the force and motion derived from heat are modified and transmitted by a pumping engine in doing the work of pumping water. And, viewed broadly as a machine doing work, the pumping engine may be divided into two principal classes of parts; the framing and bedplates, or non-moving or structural parts, and the mechanism or working or moving parts. The frames and bedplates provide support for the moving parts and to a considerable extent control their movements.

In the operation of a pumping engine, the things that are done or accomplished are:

a. A natural source of energy, as, for example, heat, gives out force and communicates action or movement to the working parts.

b. The force and motion so derived from the source of energy are transmitted through the piston and plunger rods to the water plungers within the pumps, in a proper and suitable manner and amount, to do the work for which the engine is designed.

c. The water plungers, or the real working members of the machine, by means of their force and action together, pump the water against the pressure and in the quantity for which the pumping engine is constructed.

The strength of the materials, the most suitable materials to be used, the manner of their use, and the necessary dimensions, in order that risk of breakage and overstraining may be avoided so far as practicable in the regular work of the machine, while undergoing the greatest straining action called

for by the work being done, are to be carefully determined in order that a pumping engine may be fit for use in regular water works service for long periods of time.

The proper materials, then, for the various classes and parts of pumping engines having been selected, the work of construction consists in forming and shaping the various parts to the dimensions and plans of the design, by means of proper processes, tools, and machinery, and then fitting and securing the various parts together so as to form a complete machine ready for work. And, as this chapter is devoted to frames and bedplates, those parts principally will be dealt with at present.

The framing of horizontal and of vertical pumping engines differ very radically from each other in form and use. In a horizontal machine the steam cylinders and water cylinders are supported directly from the foundations without any assistance from the frames; the frames acting only to hold the steam and water ends of the engine in proper relation with each other so as to give the necessary amount of resistance to the working forces, and a correct design will bring the work of the engine and the strains on the frames into the same center lines and thus make the machine self contained.

The definition of the word "Science" is "Knowledge gained and verified by exact obvervation and correct thinking, or, obtained individually by study of facts, principles, causes, etc.," and engine framing certainly requires just such treatment if it is to be put in the right place and made in the proper form and strength.

But the aim of the engineer in designing pumping engines is not to be precisely and exactly scientific to the exact limits of the conditions imposed, but rather to make his work so it shall be SAFE, as a first consideration, and then go as nearly to the exact line of requirements as his experience finally teaches him he can scientifically reach, with all of the conditions weighed and accounted for. This process has been put into one short sentence by a very eminent engineer and

designer in the pumping engine line; "if it is too strong nobody knows it, but if it is too weak everybody knows it."

In the beginning, when pumping engines were built mostly for the purpose of removing water from mines, the framing did not amount to very much of itself, and in fact the walls of the building covering the working beam were sometimes used really as framing. The Simpson beam pumping engine was constructed with complete frames and bedplates, and for several years after its introduction, from 1848 to about 1880, it seemed to be necessary to build large pumping engines upon the beam and fly wheel plan, in some form or other, and quite a number of special designs embodying various applications of framing and bedplates were introduced, but not sufficiently repeated to comprise a type. In the meantime the Worthington duplex horizontal pumping engine for water works, the Holly quadruplex engine, and the Gaskill engine, had been designed, repeated, and finally took place in the regular field for such work in typical form. The older form of the Worthington duplex engine and of the Gaskill engine, both horizontal machines and types widely known, had for the framing between the steam and water ends, polished round bars of iron or steel; but the latter engine had also a substantial bedplate extending beneath each side of the machine which no doubt greatly stiffened and strengthened the engine, and was probably necessary in a crank and fly wheel machine with such light weight connecting members. The old fashioned non-rotative machine on the other hand, with its entire absence of heavy revolving or vibrating parts, no doubt made out well enough with the round bar framing. At all events, most of them have lasted many years in good order. In the original Gaskill engine there was also a cast iron girder extending from the high pressure cylinder at each side of the engine to the middle of the length of each water cylinder, at the top of the force chambers, to which were attached the main pillow blocks for the crank shaft.

The framing of the later Gaskill, and of the high duty Worth-

ington engines, are composed of comparatively heavy and certainly very strong cast iron girders; in the use of which in the former engine the bedplates have been discarded and both steam and water cylinders set directly and secured upon the tops of the foundation piers.

The framing of the Holly quadruplex engine was in the "A"-form, with the main crank shaft at the top of the "A" and the center lines of the steam and water cylinders at an angle; the main pumps were located immediately behind and below the steam cylinders, with struts or distance pieces to keep the water and steam cylinders in correct alignment and distance.

Fig. 102, Fig. 103, and Fig. 104 show the framing of the Holly quadruplex, and the later frames of the Worthington and Gaskill engines.

In quite recent years a design of crank and fly wheel pumping engine known as the horizontal cross compound has come to the front as a pronounced type, although cross compound steam engines and an occasional cross compound pumping engine have been known for a long time. An attempt was made a good many years ago to produce the Worthington engine in the form of the cross compound, and called the tank engine from the fact that what is now known as the receiver, was known by the name of the tank, or at least so designated by the builders of the non-rotative machine. Several of these tank engines were constructed, but the delicacy of balance between the two sides of the machine necessary for good results in the crankless engine, could not be obtained where the steam was entirely let go of by one side and taken up again by the other side. In a crank engine, the two sides of the machine are held at all times in proper relation with each other by means of the absolute control of the moving parts by the cranks, and a little surplus power by one side or the other did not matter, as the shaft and wheel took care of all irregularities when they occurred.

The Allis-Chalmers Company, the Platt Iron Works Company, and the Snow Steam Pump Company have brought out

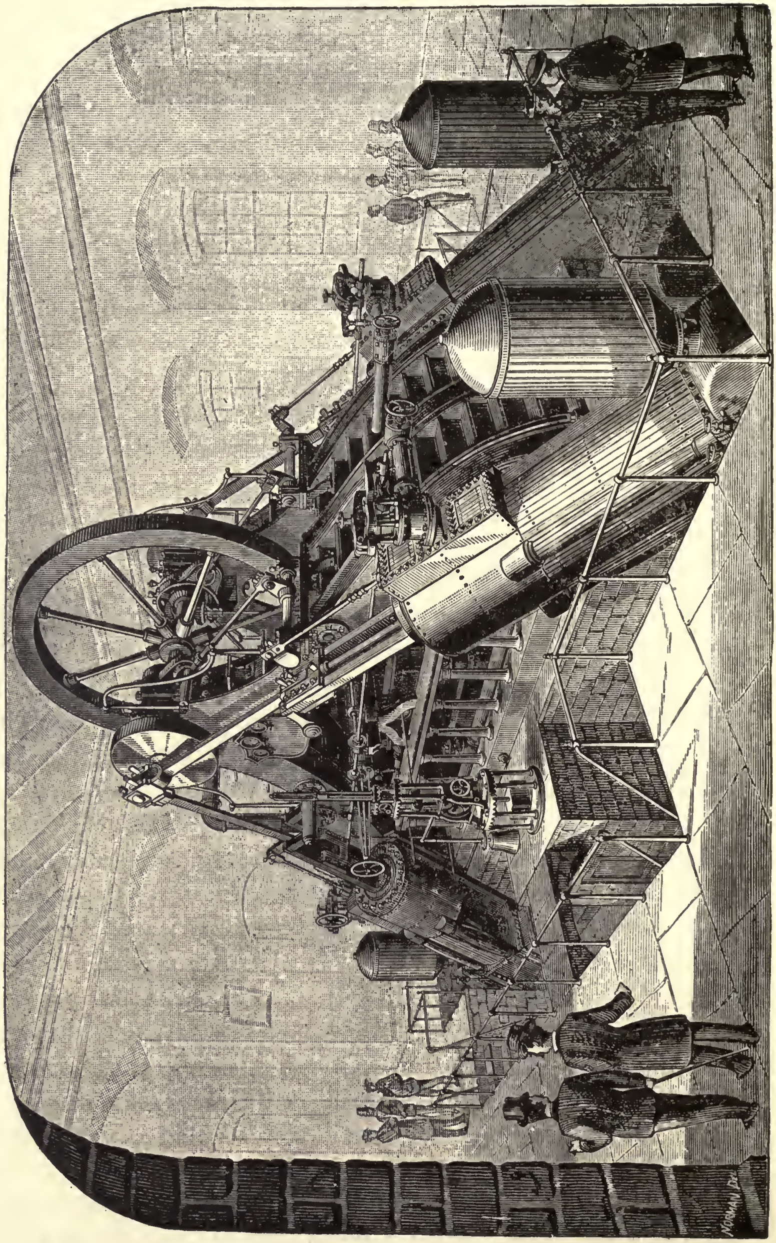


Fig. 102. — Framing of Holly Quadruplex Pumping Engine.

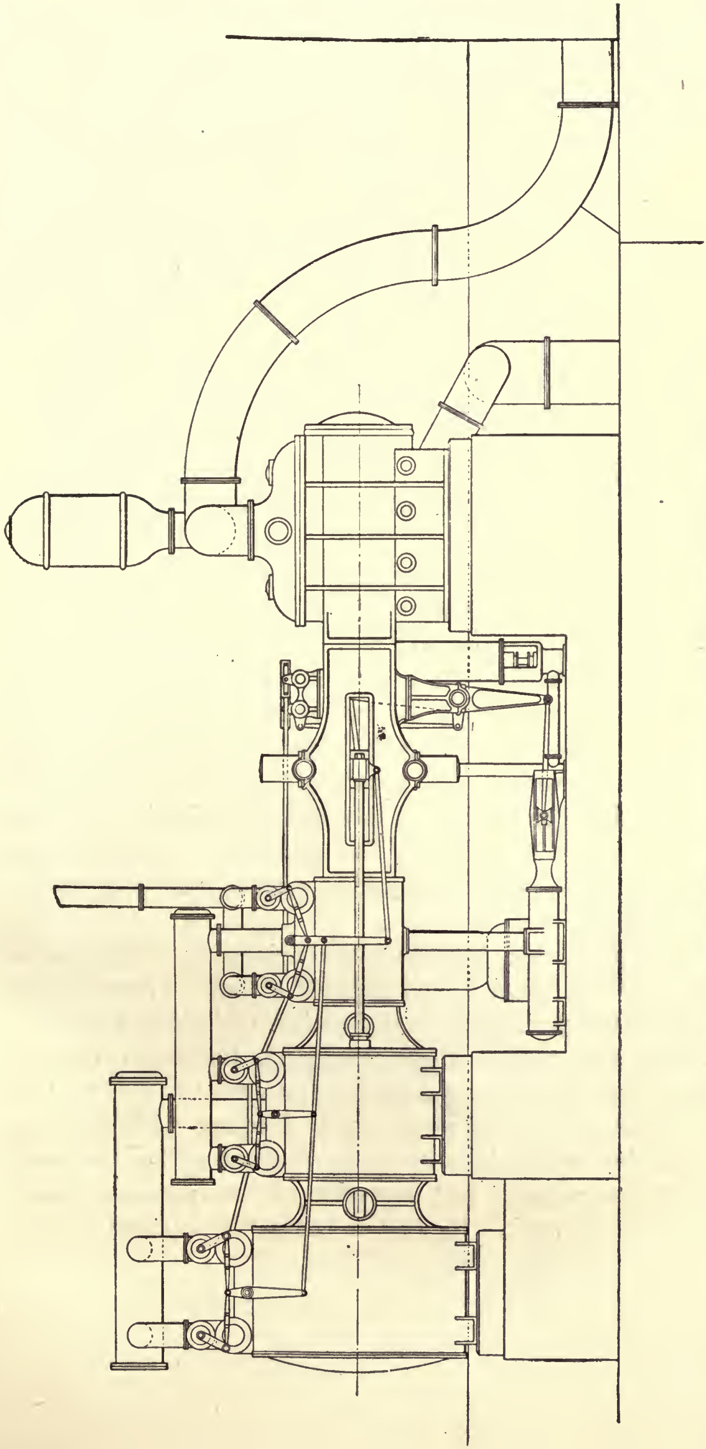


Fig. 103. — Framing of Later Worthington Pumping Engine.

very effective and substantial machines of the cross compound type, and they consist really of cross compound Corliss steam engines, connected to two water cylinders by means of very heavy framing secured to the foundation by means of anchor bolts. The main pumps in the Allis-Chalmers and Snow engines are situated at opposite ends of the frames from the steam cylinders, and in direct line with them. The water cylinders and the steam cylinders are firmly bolted directly to these engine frames of a deep box pattern; and a direct and rigid driving connection is made between the steam pistons and the water plungers by means of heavy steel distance rods, which connect piston and plunger cross heads at opposite ends of the framing; the cross heads at the steam end being coupled to the connecting rods for driving the cranks, which are pressed and keyed on to the ends of the main shaft at an angle of 90 degrees.

The cross head guides in all three of these engines are formed in the frames in a very solid, effective manner; and the main pillow blocks are also cast as a portion of the frames where they are broadened out for their accommodation. The fly wheel is located mid-way between the two main frames, and in the Allis-Chalmers and Snow engines, about in the middle of the machine, both crosswise and lengthwise. This type of pumping engine requires considerable floor space in proportion to capacity, but it is a very effective and satisfactory machine, combining as it does a reasonably high economy of steam with a moderate price.

Fig. 105 shows the Allis-Chalmers engine, Fig. 106 shows the Platt Iron Works engine, and Fig. 107 shows the Snow engine, illustrating the peculiar form of framing, and other details.

This present typical cross compound, horizontal pumping engine, is no doubt a very good investment for pumping, up to a few hundred horse power; and of course with the horse power limited, which it must be on account of the horizontal cylinders, its capacity will depend upon the pressure against which it is to work. Its cost is comparatively low, and its

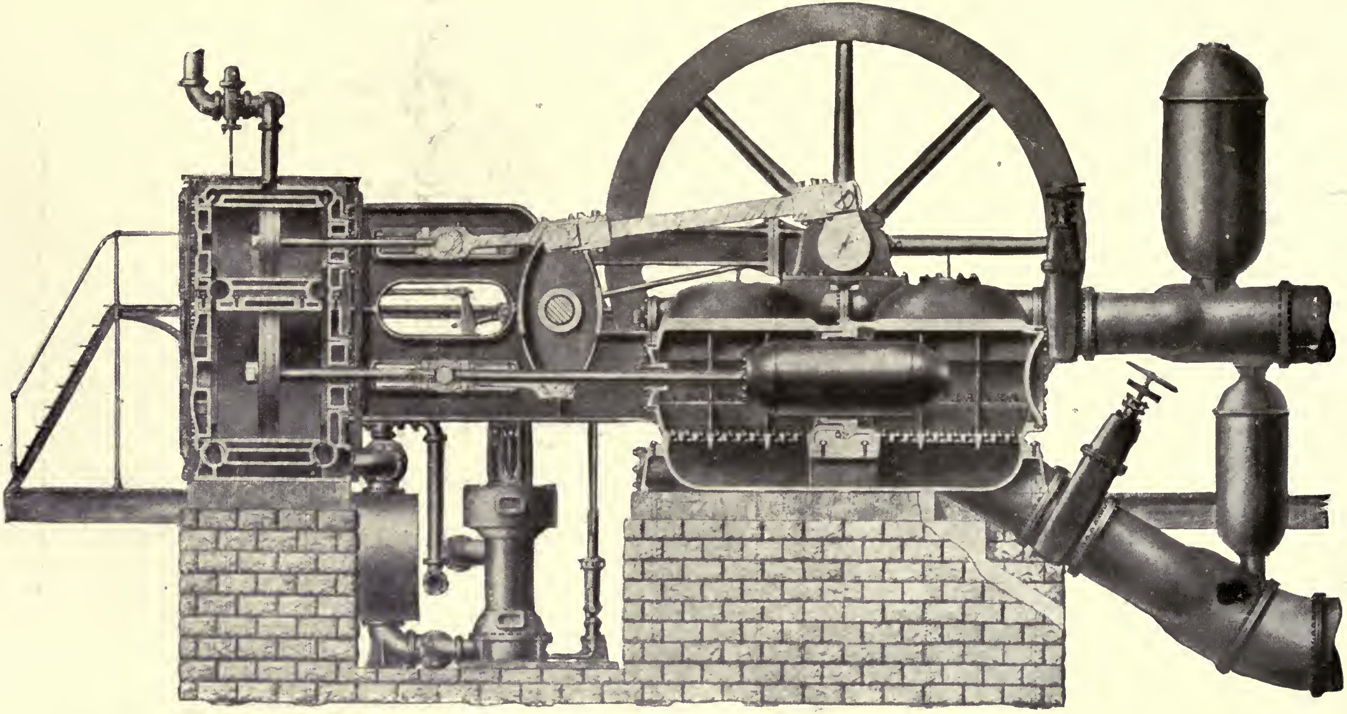


Fig. 104. — Framing of Later Gaskill Pumping Engine.

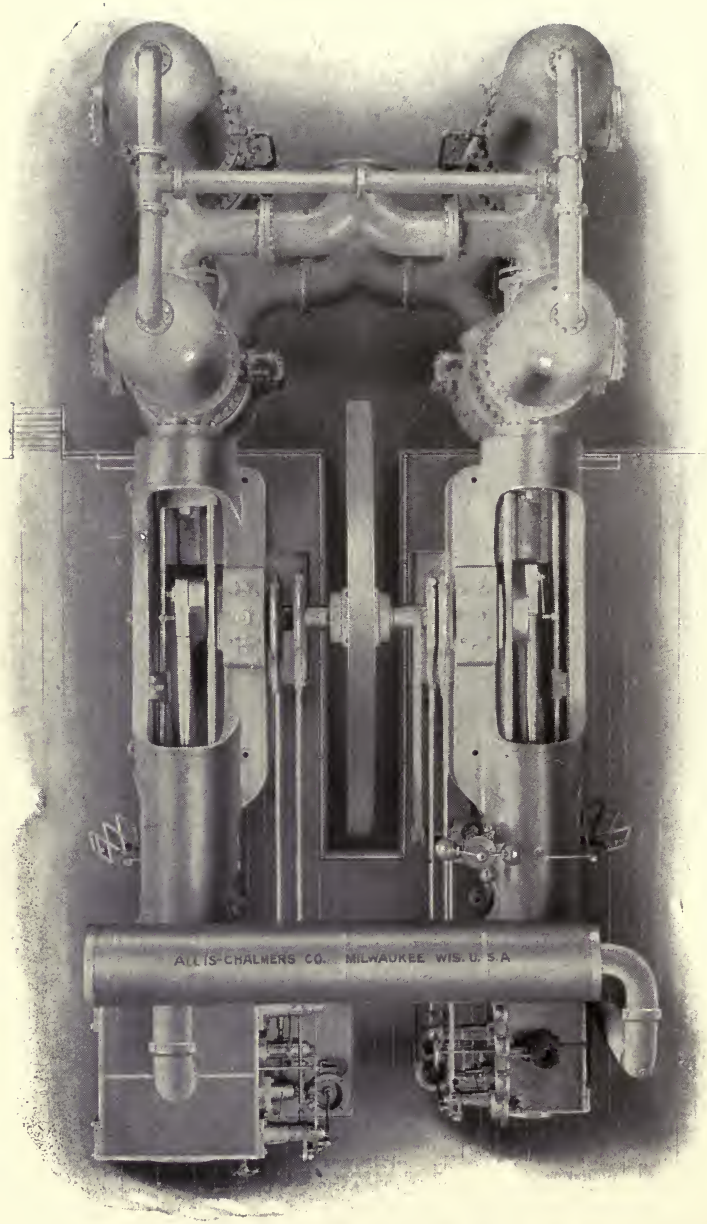


Fig. 105. — Allis-Chalmers Cross Compound Framing.

steam duty is comparatively high; in fact, its duty can be made about as high as any compound by introducing into its construction some of the refinements applied to the vertical triple expansion machine, as in the case of the Platt Iron Works

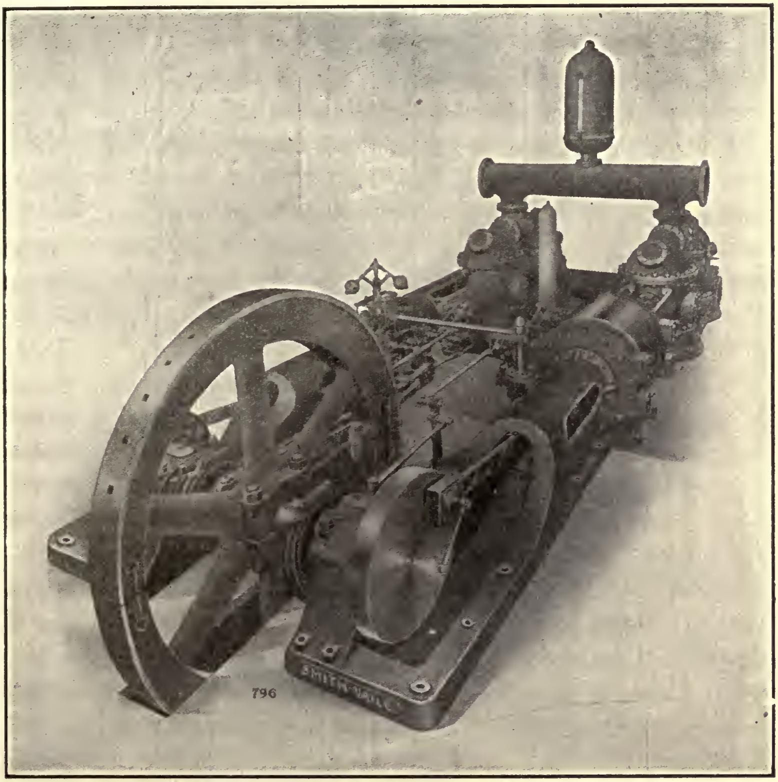


Fig. 106.— Platt Iron Works Cross Compound Framing.

engine, and under the average water works head of, say, 200 ft., the cross compound machine might be favored up to 10,000,000 U. S. gallons capacity. On the direct service, or on the standpipe system, where there is no storage by the use of reservoirs, by calculating the maximum capacity of the engine at a fairly good piston speed, but where it will likely run at not much

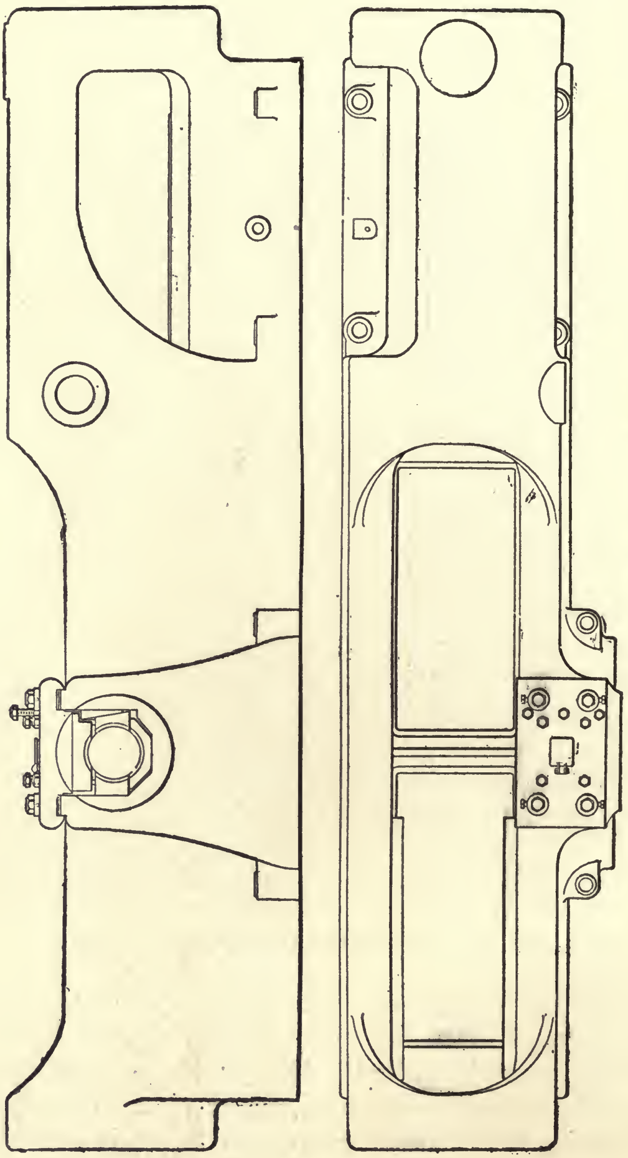


Fig. 107.—Snow Cross Compound Framing.

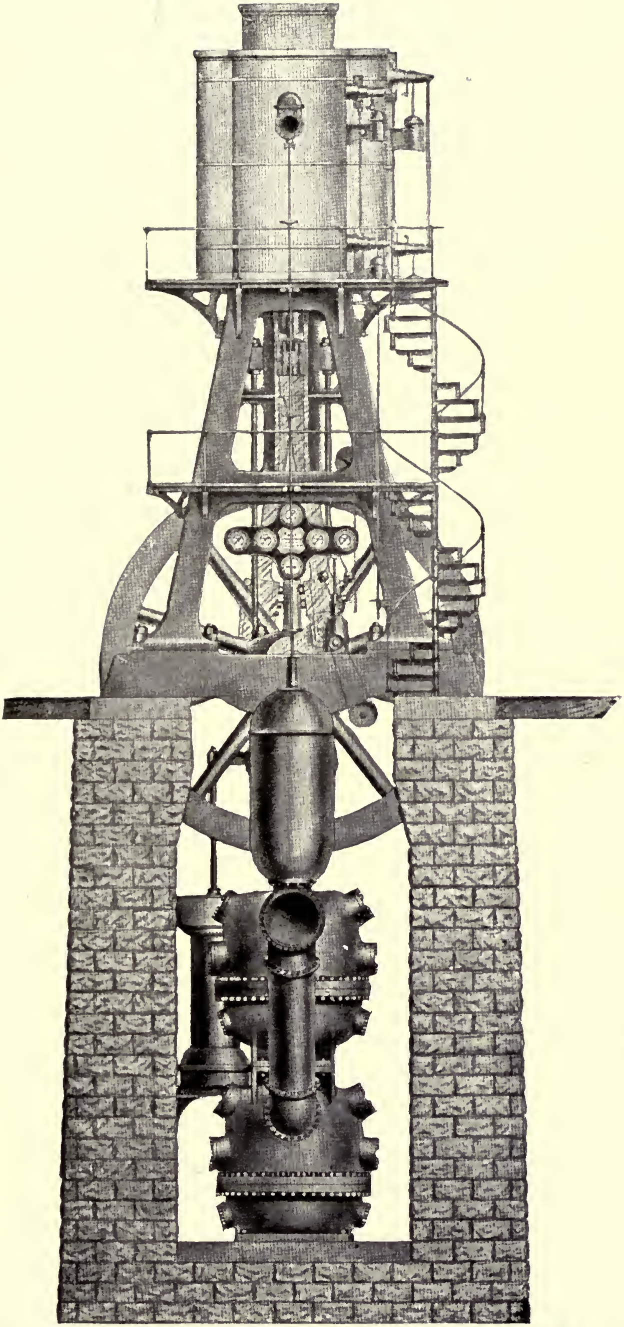


Fig. 108 — Vertical Pumping Engine with two Piers, Holly Mfg. Co.

more than half speed most of the time, a satisfactory capital economy and a very good general steam economy can be effected.

With vertical pumping engines the framing is, as may be supposed, entirely different from that of the horizontal machine. In the early days of the present type of vertical pumping machinery, say twenty years ago, the pumps were placed in a pit formed by the massive stone or brick piers which made up the foundations for supporting the steam end bedplates, and upon the tops of these bedplates the framing, mostly of the "A" frame type, was securely bolted. Of the crank and fly wheel engines, a very few at the beginning had their steam cylinders located upon the bedplate, and the pillow blocks for the crank shaft at the top or apex of the "A" frame, but this was soon changed, and what is really the modern marine engine with the steam cylinders at the top of the "A" frame and the pillow blocks and shaft at the bedplate level, soon came to the front, and the present vertical engine likely will remain so until it is supplanted by some other form of prime mover, not yet in sight.

After a few vertical pumping engines had been built and put into operation, it was found that one of the masonry piers could be dispensed with, and one end of the main bedplates could be supported on the tops of the pump chambers, thus making the machine partly self-contained; and this change also made everything about the main pumps more accessible. Later on still another change was made, and the entire machine was made self-contained, by using "A" frames extending from sole plates at the bottom of the structure, and upon these sole plates were bolted the main pumps. The frames, just about where the cross bar of a capital letter "A" would be located, are intercepted by the steam end bedplates; then the frames extend upward until the rather broad apex or top of the "A" is reached, where the steam cylinders are located.

Fig. 108 and Fig. 109 show the vertical crank engine, with two masonry piers, and also with one pier. Fig. 110 shows

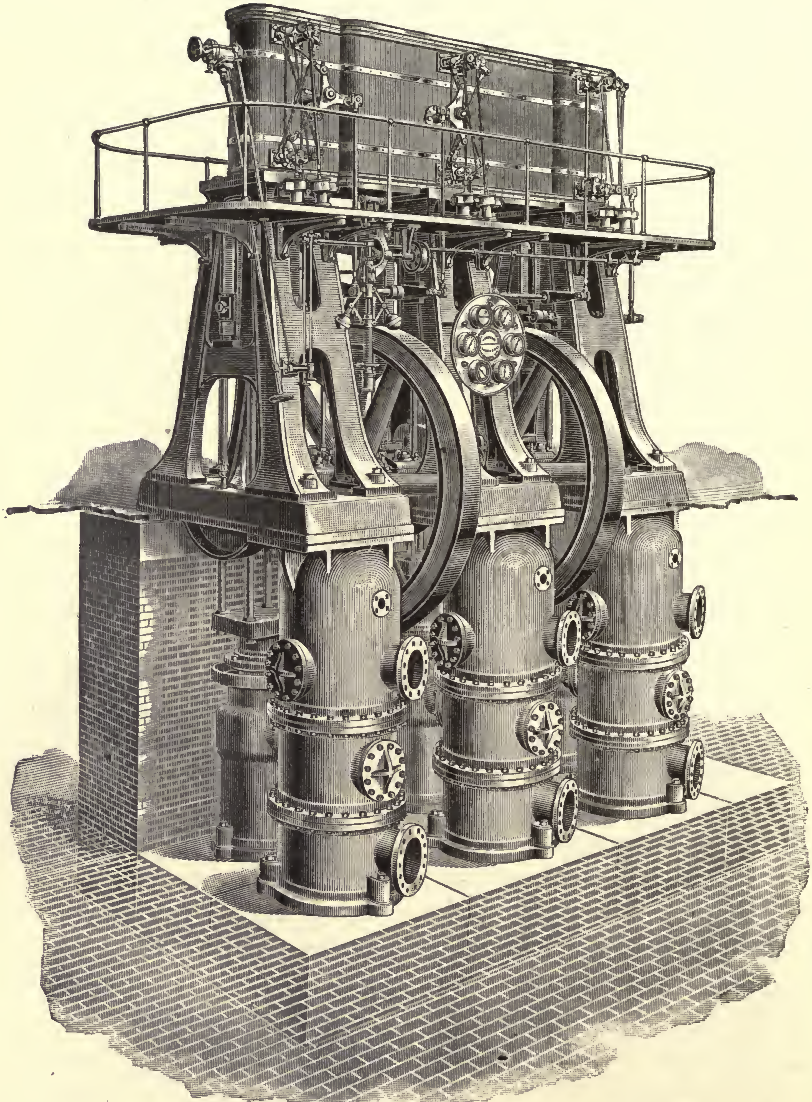


Fig. 109. — Vertical Pumping Engine with one Pier.

the vertical engine with the "A" frames extending from the bottom of the water end to the steam cylinders.

So far, or up to about eight years ago, what is known as the

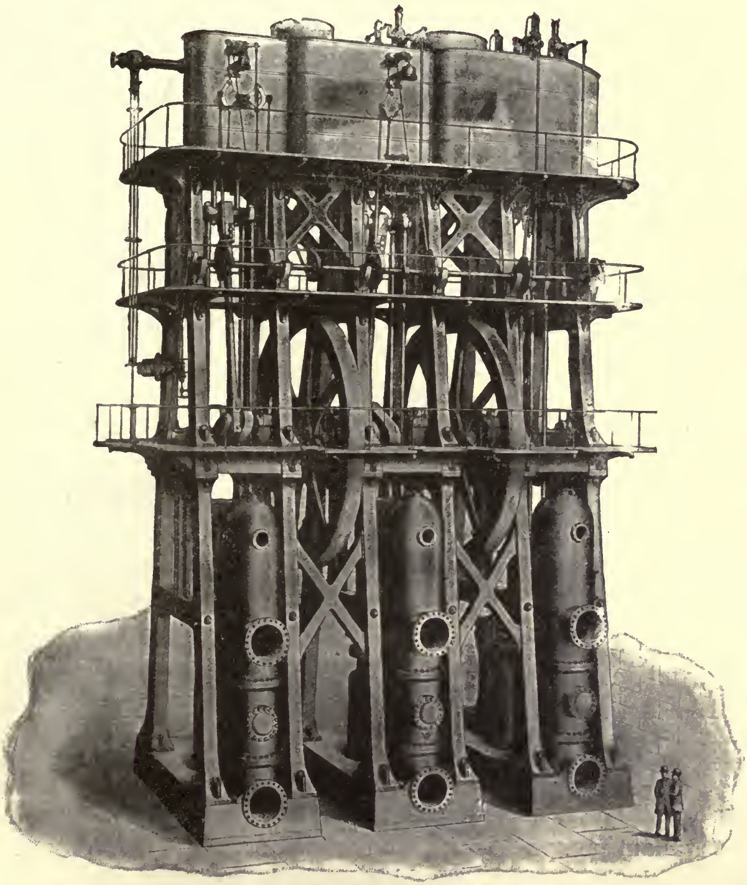


Fig. 110. — Vertical Pumping Engine with all "A" Frame.

double "A" frame had been used, but about that time a still nearer approach to the later marine construction was made, and the double "A" frame gave way to the single "A" frame, which maintained all of the principal constructive effects but made a

much neater and in some ways a more accessible engine frame. The adoption of the single frame made it rather awkward to use a double frame below the steam bedplates, and so another change, which looks as though it might be a final one, is to support the entire steam part of the machine from the steam bedplates up, on top of the valve or air chambers of the main pumps. The main pumps are supported upon sole plates in the basement of the engine room, and for all ordinary depths of basement possible to obtain in pumping stations this type of pumping engine will answer.

This construction, as a whole, is very effective and substantial, neat and symmetrical. And so far as can now be seen, this self-contained, solidly appearing, and really very solid type of pumping engine of the triple expansion class is quite the limit of perfect design and construction of crank and fly wheel machinery for large engines.

Fig. 111 is a good representative illustration of this machine, and aside from admirable arrangement of bedplate and frames the various other details may be seen to be most fittingly adapted in the general design. There are no foundations required excepting a level and substantial bottom in the engine house basement, resting upon rock preferably, but any good soil with the pressures properly sustained, using piling if necessary, will answer the purpose.

This design is also very satisfactory as a cross compound, vertical pumping engine, by simply leaving off a third of the design, using one high pressure and one low pressure cylinder, two frames, two bedplates, one shaft and wheel, and two of the main pumps. In such a case if these lines were followed closely, the two pumps would have to be coupled with cranks set opposite each other, or set with the crank pins 180 degrees apart, on account of the single acting plungers which must act directly opposite each other. Such a cross compound would do very good service on reservoir work, where the force main had no connections with the service pipes of consumers.

The double "A" frame engine has been built also in the

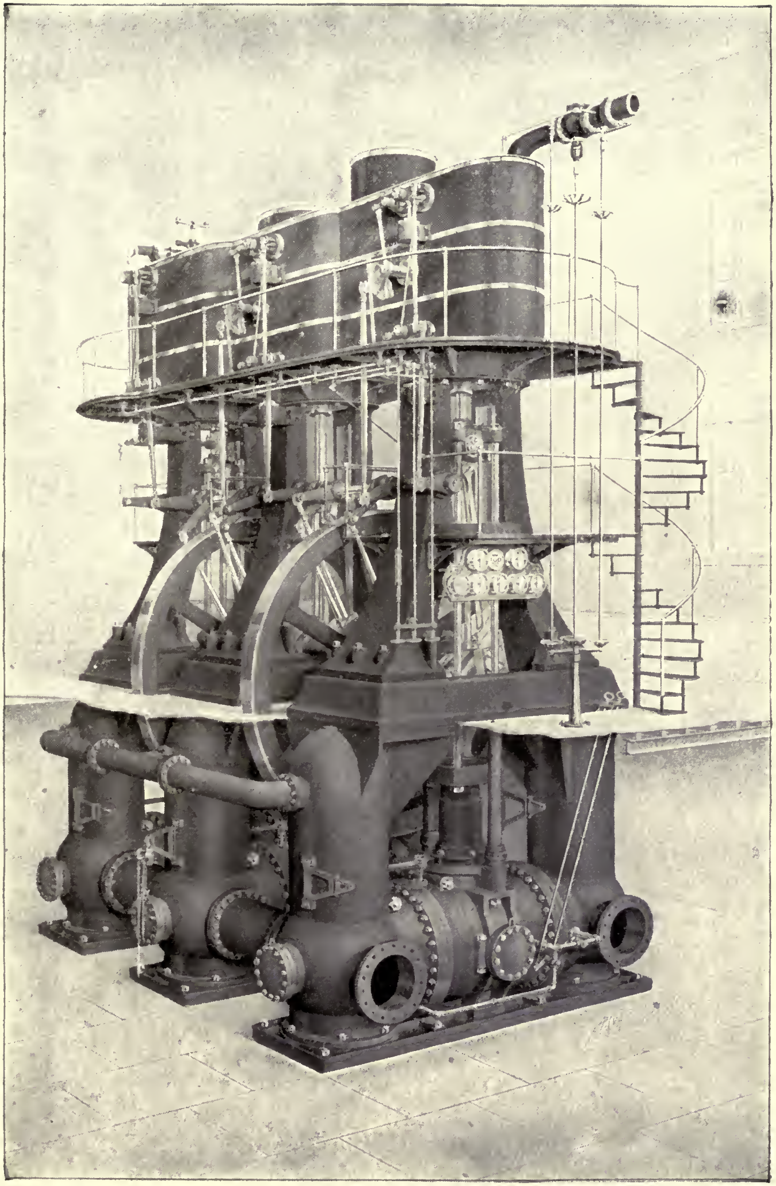


Fig. 111. — Latest Type of Self-Contained Engine, Irving H. Reynolds.

cross compound form; with the single masonry pier, and also with the completely self-contained "A" framing. In this form, small and moderate sized compound engines have been built, and also so large as to have 50 inch high pressure and 92 inch low pressure cylinders, with 64 inches stroke; also with 27 inch high pressure, 52 inch low pressure, and with 108 inches stroke.

The Worthington triple expansion, high duty vertical engines have been built as large as 20,000,000 U. S. gallons daily capacity, and a few even larger than this. In this type the framing is somewhat of the "A" form, although not so clearly pronounced as in some other types; there are generally no bedplates, but the feet of the frames are secured by heavy anchor bolts to the tops of the masonry piers between which the main pumps are located in a sort of pit formed between the piers. These piers rest upon a sub-foundation usually of concrete upon which the main pumps are secured; thus the engine piers really form a portion of the framing.

In a late design, the entire engine is placed upon a sole plate with "A" frames extending from the sole plate to the steam cylinders at the top of the construction, and so the machine is completely self-contained, in this respect differing from the first verticals of this type.

It is curious to note in this connection how several of the makers of large pumping engines have been mistaken at different times and unknown to each other, regarding the needs of a rigid frame connection between the steam and water ends; and the writer recalls curiously enough also, that in his own experience he observed and assisted in adjusting this error in two different widely separated cases, involving two of the largest builders of pumping machinery at the different times. The first was a case of three single acting, outside packed plungers located in independent plunger barrels at the bottom of a masonry pit formed by the foundations. It had been calculated that these plunger barrels were to be placed upon solid stone masonry at the bottom of the foundations;

and that as the thrust of discharge was always downward, and that the suction lift was almost nothing when compared with the weight of the plunger barrels, to say nothing of the fact that the plunger barrels were heavily bolted to the valve chambers, making a great mass of iron castings weighing many tons. But there was no frame connection from the plunger barrels to the steam end bedplates at the top of the foundations, it being presumed that none would be needed. Nevertheless with all appearances in favor of satisfactory action, there was enough relief upon the stonework under the plunger barrels during the suction stroke, to cause a decided downward thrust during the delivery stroke, and enough vertical action took place to begin to wear the masonry seat of the plunger barrels, promising in a short time to throw the machinery out of line by unevenness of the foundations. The remedy applied was to put connecting struts between the steam bedplates at the top of the pit and the plunger barrels at the bottom, and so placing something to resist the working forces in the center plane of motion.

The other case was a very large vertical non-rotative pumping engine, with two double acting plunger pumps at the bottom of the pit formed by the foundation piers, the steam end of the machine being supported upon "A" frames from the tops of these piers, and without any framing connection between the steam and water ends. It had been calculated that the immense weight of the water ends would make all such connections unnecessary, but a short time at work demonstrated that the concrete foundation beneath the main pumps was beginning to give way, and it finally had to be replaced, and reinforced by heavy steel " I " beams. Therefore it would seem that as an absolute principle of construction, in pumping engines at least, and probably in all prime movers, there must be a rigid frame connection between the power and the work, central with the lines of force.

Fig. 112 shows the Worthington vertical triple expansion pumping engine, and indicates the arrangement of the fram-

ing, and the positions of the steam and water ends of the machine.

It being necessary in the non-rotative type to have a pumping engine which will make four strokes to a "revolution" or cycle, the main pumps have to be double acting, and the upward discharge stroke exerts a very considerable upward pressure upon

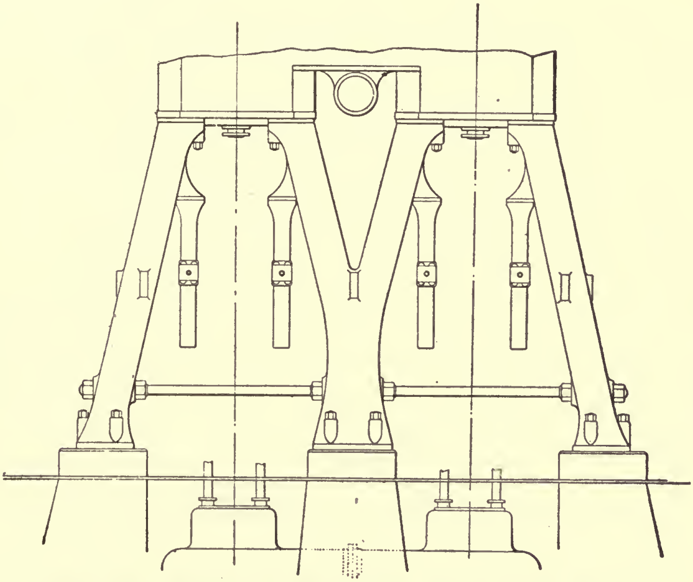


Fig. 112.—Framing of Worthington Vertical Triple Engine.

the main pump casting, in this case a great many tons, which although rather a long way short of the weight of the water cylinder, is nevertheless quite sufficient to cause an upward and downward vibration. The action of such a performance would resemble somewhat the treading of an enormous cast iron elephant, weighing perhaps 50 or 60 tons; and any one at all conversant with concrete and masonry, would know how damaging it would likely be. The placing of struts and tie bolts between the steam and water ends in the line of the work,

would make the machine self-contained and avoid such bad effects upon the foundation.

In connection with this branch of the subject, there naturally comes the matter of galleries, platforms, stairways, and more or less complete facilities for getting around the engine, at rest and in motion. There should be a substantial and permanent iron platform around and about the upper ends of the main plunger barrels for vertical engines, at a convenient height for easily and safely reaching the gland bolts and plunger packing, when necessary to renew or take up the packing.

The main floor of the engine room will provide means for manipulating the throttle valve, condenser valves, priming valves, and other necessary facilities for starting and running the engine. Then a platform at the base of the steam cylinders on moderate sized engines, and in addition to this a platform half way up the frames on large engines; also on the larger machines there is need of a platform somewhere near the tops of the steam cylinders.

With horizontal pumping engines there is no need of so many platforms, but some are necessary for the larger sizes; and with large Gaskill and Worthington engines, especially for the former, a platform extending the entire length on both sides and back of the steam cylinders, located about at the base of the high pressure cylinder, is very useful and convenient for those who have to be about the engine. With the Worthington and large cross compound engines, a bridge platform across the machine, and facilities for reaching the valve stems, gauges, and other details are usually provided.

CHAPTER XXV

MATERIAL FOR PUMPING ENGINES

In this age and day when skilled practice of wide range, and good business management, for foundry, forge, and machine shop, have been brought to a very high stage of perfection, the price of a high class economical pumping engine has been reduced to a point which justifies the use of a very high steam economy in pumping water. A much better engine can be installed now than most people would buy a few years ago; an engine in which is combined simplicity, high efficiency, great durability, and compactness. In fact, a high type of pumping engine is now used in the comparatively small capacities, which, not so very long ago were left to the field of low economy and low interest account. It may be that profits are less; but production is certainly less costly; several factors no doubt conspiring to help the buyer.

In no direction probably has there been so much improvement in the line of material and workmanship as in the crank shaft and similar details of rotative engines. For use in the construction of crank and fly wheel pumping engines the economical production of the very highest quality of solid open hearth steel forgings, and center forged steel details for steam machinery has gradually brought the grade of various parts of pumping engines up to the highest known level for such work. Although costing a moderate percentage more than wrought iron, the very best and really cheapest material for piston rods, connecting rods, crank shafts, distance rods, and other details of like nature, is steel; forged under hydraulic perssure from clean, sound ingots somewhat larger in mass than the finished product, and the piece properly annealed. Such

forgings not infrequently have a tensile strength of over 80,000 lbs., and an elastic limit of 50,000 lbs. with an elongation of 25 per cent before parting.

Open hearth furnaces with temperatures of about 4,000 F. are used for steel ingots necessary for making such shafts, rods, etc. The best forging is mostly done in hydraulic presses at quite a moderate rate of "squeezing" which allows the flow and adjustment of the metal to take place. Sometimes gross pressures, of from 12,000 to 14,000 tons, are employed in this hydraulic forging process, and from 400,000,000 to 600,000,000 foot pounds per minute are developed; the actual force in the pressure pumps reaching as high as 6,500 lbs. per square inch. During the process of forging a hollow steel shaft inside and out, the material is thoroughly compacted, as the mandril almost fills the hole in the ingot and serves as a sort of anvil, a succession of pressures taking the place of the blows in ordinary steam hammer forging; the process being naturally assisted by the advantages in heating a hollow cylindrical mass instead of a solid one. The undesirable qualities in the fluid compressed steel ingot are in the center, and after the original casting of the ingot is made the defects are bored out after cooling, and the sound piece in the form of a short thick cylinder is reheated for forging; so that the material which is actually worked upon for the finished product is practically homogeneous, or alike in quality throughout.

Solid forgings are often used instead of the hollow forgings described above, and very good material may be produced, especially of the dimensions used in pumping engines. Some work of this character for large pumping engines was tested by the writer during the past year with the following results; the records here given are taken at random from a number:

Designation of Specimen.	Square Inch, Elastic Limit, Pounds.	Square Inch, Tensile Strength, Pounds.	Elongation, Per Cent.	Reduction of Area.
A	35,300	76,250	25	39
B	33,150	77,950	27	37
C	33,250	75,350	27	37

The cranks are also very important members of the construction and are made of cast iron, wrought iron, or steel, mostly the latter in the later and better machines, according to circumstances and the ideas of the designer. Cranks are sometimes forced on to the shaft by hydraulic pressure or in a screw press, and are sometimes shrunk on. The fly wheel hubs are also subjected to the same treatment, mostly forced on when the wheel is made with a center or hub instead of in halves as is often done. Wheels in halves are of course admissible, but are not nearly so good as the built up wheel with a solid hub put into place upon the shaft before the cranks are forced into place. The fly wheel built up in sections and bolted between the two discs forming the hub by through bolts having a driving fit in reamed holes is the best work in this line; the rim is made in sections, generally eight, one arm bolted into the hub for each section, and the sections of the rim are generally held together at their ends by means of forged links set into sockets at the sides of the wheel rim, the links having been heated and allowed to cool and shrink in place. This form costs the most of any, the wheel made in halves being the other extreme as to cost and desirability; the wheel in halves is a fairly good one when well made, especially if the place of parting is planed and bolted instead of being cast as one wheel and then broken in halves. A very good compromise between these two wheels, and costing somewhere between the two types, is to have the solid hub forced on to the shaft with discs separated for the arms the same as for the built up wheel, and then have the rim made in two sections like the wheel in halves, the rim secured by means of links the same as the eight section wheel; the difference being that there are four arms to a section instead of one arm.

In the best class of work, the pressing fits are generally ten tons pressure for each inch diameter of shaft or crank pin or wheel hub seat, the wheel centers, cranks and crank pins; as, for example, a shaft 16 inches diameter would require 160 tons pressure to force on a crank or wheel center; or for a crank

pin say 8 inches diameter, a pressure of 80 tons would be required; for other parts requiring forcing fits, the pressure is used according to the experience and practice in any particular line of the shop doing the work.

Regarding the dimensions of crank shafts, practice varies; but the best and strongest work in the line of crank and fly wheel pumping machinery follows rather closely the lines of power engines in relation of shaft diameter to horse power and speed. The work of the pumping engine is largely done directly from the pistons to the plungers; and in the non-rotative engines of the low duty class all of the work goes directly to the plungers; in a non-rotative high duty engine, as, for example, the Worthington, the surplus power at the beginning of the stroke goes into the compensating cylinders and through them lifts the accumulator piston, this piston in effect resembling a large pneumatic dumb bell, or a dumb bell consisting of air pressure, the dumb bell returning the power after the compensating plungers pass the central point in the stroke. Even the equalizing of the expanding steam pressure is accomplished by direct pressures and without transmission through anything like a shaft; in fact, such work is done in the non-rotative machine without torque.

In the crank and fly wheel pumping engine, however, only a portion of the power goes directly from the pistons to the plungers; the first part of the stroke, when there is a surplus of power, sending a portion of the work to the wheel to be given back again during the latter part of the stroke when the expanding steam is below the working requirements. Theoretically, therefore, as all of the work is not transmitted through the shaft, it might be naturally enough supposed that a smaller shaft than a power engine requires, would answer for a pumping engine, but in the latter the transmission of the power to and from the wheel is a rougher sort of a task than simply sending small installments of energy into a steadily revolving wheel at a pretty good speed. It may be readily noted that the dimensions of crank shafts in pumping engines

just about correspond to the result of recognized formulae for transmitting power by shafts; that is, the shafts are apparently calculated as though all of the power was sent through them; and in fact the particular work they have to do when the stubborn attributes of water are considered, is no doubt a more severe trial to the material even with only a moderate percentage of the power of the engine actually transmitted through the shaft in the form of torque or twisting strain. At all events, if in the triple expansion engine the crank shafts, of which there are two, are considered as each taking care of half the power, which is a reasonable idea to say the least, Rankine's formula for transmitting power by a shaft gives just about the diameter that experience has taught the builders will produce a shaft safe for such a machine. If the same formula be applied to a cross compound engine of a well recognized make, upon the supposition that the shaft is to be able to transmit the full power, the actual shaft made for the machine will be found to be very close in diameter to the results of the formula. But aside from what is found in practice, the Rankine formula will be found to give a safe and practical size of shaft when applied to pumping engines.

Although perhaps a little out of place in a book of this kind, the above mentioned formula is so interesting that it is given:

H.P. represents foot pounds per minute of the horse power of the engine; found by multiplying the indicated horse power of the steam cylinders by 33,000.

6.2832 is the ratio between the length of the radius of a circle and the circumference of a circle; or the proportion between the distance from the center to the outside of the shaft, and the distance around the shaft.

R.P.M. represents the revolutions which the engine is to make per minute at full contract speed.

M represents the mean or average twisting strain or stress under which the shaft is placed by the work which is done.

MM represents the maximum or greatest twisting strain or stress under which the shaft is placed by the work which is done; and is found by multiplying *M* by 12 for the reason that 6 is ordinarily taken as the factor of safety for materials of this kind for the mean or average strain, and this factor of safety 6 is doubled, making 12 for the maximum or greatest strain.

A represents the modulus of stress; which is a number that measures the actual value of the material for use; or, it is the measure of the force required to break a substance across, as compared with the force required to break a bar of the substance one inch square. This value is 225 for forged steel.

D represents the diameter of the shaft in inches.

Then the formula is:

$$\sqrt[3]{\frac{\left[\dfrac{H.P.}{6.2832 \times R.P.M.} \times 12\right]}{A}} = D$$

And this interpreted into words means as follows:

In this example the horse power is taken at 800, or 400 for each shaft of a triple expansion pumping engine; then, 400 × 33,000 = 13,200,000 ft. lbs. per minute, representing the work of the indicated horse power or the H.P. in the formula.

Then, as the revolutions per minute are 22.5, which represents the R.P.M. in the formula, this number is multiplied by 6.2832, which gives 141.372, by which the 13,200,000 already found is to be divided, and the result is 93,370 as the value of *M* in the formula.

Then, *M* is multiplied by 12, and the result is 1,120,440, which is the value of *MM* in the formula.

Then, the value of *MM* or 1,120,440 is divided by 225, the value of forged steel in the formula, and the result is 4,979.73, and this number is the cube of the diameter of the shaft in inches.

Then, extracting the cube root from 4,979.73, as this number is the cube of the diameter, we have 17.077 inches as the diameter of a steel forged shaft for this machine; or, say, 17 inches as the nearest practical machine shop size. It may be noted, that a recently built triple pumping engine of 800 horse power has a shaft at each side, 18 inches diameter for a short space at the middle where the wheel goes on, and 16 inches next to the pillow blocks, which is a rather close size for 17 inches average.

Still another triple pumping engine by a different builder of 500 horse power, or 250 for each shaft, has a steel forged shaft at each side 15 inches diameter, and the formula calls for 15.18 inches.

Also a cross compound pumping engine, recently tested by the writer for duty, running at unusual speed and under unusually high water pressure, and altogether presenting entirely different conditions from the usual practice, the horse power was 1,200 and the speed was 67 revolutions per minute. The formula calls for a shaft $\frac{1}{4}$ of an inch larger than the shaft found in the engine.

It is pretty certain that the Rankine formula was not used on any of the above mentioned engines, but it is very certain that experience and judgment, based upon what has been done and observed by engine builders, have established a rule for shafts which arrives at the same conclusion as was arrived at by the most eminent authority on the strength of materials. And this brings again to mind the definition of science, "Knowledge gained and verified by exact observation and correct thinking; knowledge gained individually by study of facts, principles, causes, etc.; the habit or possession of exact knowledge."

Regarding the matter of cranks and crank pins for the fly wheel pumping engines, the best are made of the forged steel; practically the same material as that for the shafts, which has already been gone into at some length. The dimensions of these parts differ some, but not very much, among different

builders. The crank pins may be set down as of a diameter and length which will produce about 1,300 lbs. per square inch of projected area, or, in other words, 1,300 lbs. for each square inch found by multiplying the diameter by the length of the journal. This would be for the outer or end crank pins of a triple machine; and the middle pin, or that for the intermediate cylinder, would be about 50 per cent greater in diameter than the outer one, or those for the high and low pressure cylinders, with the length of the journal the same in all.

The dimensions of the cranks vary some with the different builders, and even the same builder is not always consistent among different sizes and powers of engines; the convenience of manufacture is often considered first where a departure from some standard proportion does not do any harm; and on account of convenience or economical handling of the work, sometimes a heavier crank may be put on than would be calculated from the exact proportion of the machine.

However, a very good proportion for steel forged cranks would be as follows:

Main hub of crank, twice the diameter of main journal.
Thickness of hub, to be 0.65 of diameter of main journal.
Crank pin head of the crank, twice the diameter of crank pin journal.
Thickness of crank pin head of crank, 1.125 times the length of the crank pin journal.
Thickness of crank arm, 1.125 times the length of crank pin journal.

Regarding cross sections of the connecting rods, there is also considerable variation, but a very good rule is as follows:

Tensile strain at neck of connecting rod not more than 2,500 lbs. per square inch.
Tensile strain through the smallest section of head or strap 2,200 lbs. per square inch.

These strains are based upon the gross effective steam press-

ure at the beginning of the stroke, found by multiplying the area of the piston by the difference in pressure per square inch, between the initial and counter pressures. In any multiple cylinder engine the total net pressure would not be alike in all of the cylinders, so the diameter of the rod and the dimensions of the strap could be based upon the greatest strain given by any one cylinder, and then the other rods and straps made to match the largest one.

In connection with the main shafts, the main pillow blocks come in for a share of attention in this chapter; the diameter and length of the bore, the form of construction to meet different conditions, and the various details which go to make up a suitable and satisfactory bearing for the main shaft, require careful attention and a clear understanding of the requirements.

In non-rotative pumping engines of the low duty class, of course no bearings are required beyond those for rock shafts, valve motion connections, and air pump levers. In the high duty class of the non-rotative type of machinery, some require substantial pillow blocks for the trunnions of oscillating or compensating cylinders, but, take it all together, the pillow block question is not a very serious one with the non-rotative pumping machinery.

In the better forms of horizontal cross compound pumping engines from 3,000,000 U. S. gallons capacity per 24 hours, up to, say, 10,000,000 gallons, although this type seldom goes so high in capacity, the main pillow blocks are arranged so as to take up wear both horizontally and vertically, because the thrust of the engine is horizontal, and the wear from the weight of the fly wheel, shaft, and cranks, is downward. A general description of a satisfactory bearing for this type of machine is as follows:

The main frame of the engine, which is of the box form, proportionately deep and heavy, extends from the steam cylinder to the water cylinder at each side of the machine. At the inner side of the frame, or the side next to the opposite frame, the pillow block pedestal is formed as a part of the frame itself,

and at its top is located the socket or jaw for holding the boxes or shells of the main shaft bearing (see Fig. 107). The jaw and the shells, a top and a bottom and two side shells for each bearing, are generally made after the pattern of the original Corliss pillow block first made fifty years ago, and probably a better or more practical bearing was never made for a shaft. The top of the jaw is planed with projections which engage with the gib ends of the pillow block cap, thus firmly holding the two sides in proper relation; the shells in which the lining metal is poured, hammered, and bored to suit the shaft, are independent of the jaw and the cap, and are free to move horizontally when the lost motion is taken up by means of a wedge shown back of the quarter box towards the steam end of the frame. The shells can also swing if necessary just a trifle, and the general result is that these shells, having once been bored and properly fitted to the shaft, will adjust themselves to the journal in such a way as to practically avoid all tendency to bind the journal; in fact, will follow up changing positions of the journal so as to keep in line with it at all times, in such a manner that the bearing or the shaft cannot be heated or cut unless set up too tightly or left too long without lubrication. This bearing can be taken up vertically and horizontally at the same time, or each way separately without interfering with the other adjustment.

In vertical pumping engines both large and small, it is now the general practice to cast the lower half of the main pillow block in the engine bedplate, and then cover it with a heavy cap formed so as to tie together the ends of the lower half by means of lips or gibs formed at the lower outer edges of the scap. The actual bearing boxes or shells in which the journal of the shaft rests and revolves, are mostly two separate and removable pieces from the main part of the pillow block; although in some cases the lower half is prepared for the shaft solidly in the bedplate, and the upper half or cap is made to form the shell for the upper part of the bearing. In either case, the removable shells or the non-removable bearing, the real

bearing surface for the shaft is made of some sort of anti-friction metal, practically an extra good mixture of what has been long known as babbitt metal. This comparatively soft metal is melted and poured into spaces formed for it, thus making a circular surface approximately the size and shape of the shaft journal. After cooling, this lining metal is thoroughly hammered and condensed in its place and then accurately bored to suit the shaft. A system of grooves are cut in the surface of the finished bearing to distribute the oil under, over, and about the journal. The arranging and cutting of

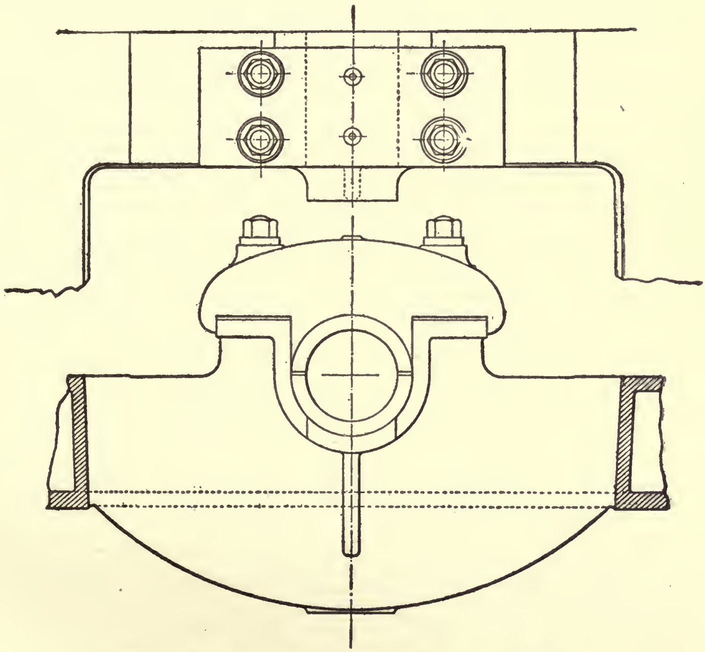


Fig. 113. — Main Pillow Block for Crank Shaft.

these oil grooves, although seemingly a minor detail, is, like a great many minor details, extremely important to the success and economy of the machine.

In vertical engines of course is there is no need of taking up the wear and lost motion in more than one direction, which

is vertical; as the thrust of the engine and the weight supported by the bearings act practically in vertical lines only. The bolting of the pillow block, the distribution of the strains in the cap, and the facilities for removing the shells, especially the under shell, so as to make as little delay as possible in the working of the engine, are all matters which, though comparatively small in themselves, are relatively of enough significance to justify the most careful attention.

Fig. 113 shows a side view and plan of a pillow block for a vertical pumping engine formed within the bedplate, and in this machine the bottom of the pillow block where the lower shell rests is made circular, so as to roll out the shell by simply taking the weight of the shaft upon a jack screw or by other suitable means. This side view also shows the strong reinforcement in the bedplate casting where the depression is formed for the pillow block.

In making a brief and general summary of the different kinds of materials which enter into the construction of pumping engines, it goes without saying that they should be of the very best quality practicable to obtain. It is simply impossible to obtain absolutely perfect iron castings, but there is a practical perfection possible to obtain from a well regulated foundry, and engineers and inspectors should have sufficient judgment and courage to arrive at a fair and proper conclusion where the integrity of a casting has been called into question. Where even heavy and large castings, the loss of which is serious to the manufacturer, have some part or parts so defective that they cannot be properly remedied, they should be rejected by the buyer or his representative. But where there are blemishes only, or minor defects that add nothing to the risk of using the casting, and which can be removed or remedied by the manufacturer, there should be no hesitation in accepting such a casting.

The writer recalls to mind a large plunger barrel weighing about 8 tons, for a pumping engine, where the lower flange was so defective from "blow" or from some small portion of

the mold having been dislodged, that it was weakened and defaced to an extent that made it really impossible to use. A great remonstrance was set up by the maker, but in spite of all appeals and regrets the casting was rejected.

In another case, an air chamber belonging to the same class of machinery and weighing probably 12 tons or so, developed a bad spot in the face of the bottom flange when the facing was being done. After the roughing cut had been finished, a blemish or defect appeared in the face of the flange about 8 inches long, 2 inches wide, and 1 inch deep. As the face of the flange was about 7 inches wide and this rut extended in a circular direction, it was decided after testing and examining the surrounding metal, to have a piece of cast iron fitted and driven into the damaged place after dressing out to sound metal, and then the finishing cut for the facing work was completed. The flange was drilled for the bolt holes, and after a short time it was very difficult to locate the blemish. This casting was then accepted for use in the engine.

The cast iron for fly wheels, bedplates, frames, etc., having a tensile strength of 18,000 to 20,000 lbs. per square inch, or a cross breaking resistance of one tenth of these figures, with usual size bars, would be acceptable for pumping machinery; and for pressure sustaining parts such as air chambers, pump barrels, valve chambers, steam cylinders, etc., the tensile strength should be as high as from 22,000 to 25,000 lbs. per square inch, and with a cross breaking strength of one tenth as before mentioned.

Good steel castings have become a common and commercial article for various parts of machinery, and are acceptable for cross heads and other parts of pumping engines where formerly forgings only were used. Such castings should be properly annealed, and the material entering into them would be acceptable when the specimens showed a tensile strength of 60,000 to 65,000 lbs. per square inch, an elongation of 20 per cent and an elastic limit of 30,000 lbs.; the usual test specimens being used in these determinations.

Very little wrought iron aside from bolts is used in pumping engines, and the usual specimens should show approximately 45,000 lbs. tensile strength, 25,000 lbs. elastic limit, and 20 per cent elongation in 8 inches.

Composition metal, commonly designated brass, varies considerably in the various parts and as used by various builders. The principal quantity is utilized in the pump valve seats and appurtenances, and should be made of new metal mixed at the time of melting, for the best work, but under ordinary conditions there is not much trouble experienced with the metal usually employed.

Bearing metal, or babbitt metal as it is commonly called, varies a good deal also, but bearing metal for heavy work should be made of copper, tin, and antimony, although, like brass, there are numerous mixtures, and most reputable builders meet this requirement in acceptable fashion.

All materials for the machinery should be subject to inspection when the buyer desires, but it is a useless expense to actually provide for such testing when buying a pumping engine, as the builder will of course add all contingencies to the price asked for the machinery. Therefore in the writer's opinion it will be better to secure the option of having the testing done when desired, and charge the cost to the buyer for the actual work done and expense involved at the time such tests may really be made.

CHAPTER XXVI

DUTY TESTS OF PUMPING ENGINES

The usual and practical purpose of the duty test of a pumping engine is to find out whether or not the engine contracted for and installed, is capable of meeting the contract guarantee of efficiency and economy, made by the builder; the results as ordinarily expressed in foot pounds of duty, really show how many pounds of water the engine will raise one foot high when it consumes either 1,000 lbs. weight of steam, or 1,000,000 units of heat. It would seem as though scientific research would call for duty tests, so as to determine the relation between the heat expended and the work accomplished by the machine; but the trouble of this is, that there seems to be no one to pay the bills upon a large enough scale to be of much use, either theoretically or practically.

And therefore the slower but equally certain method is for close hard thinkers and observers to improve from time to time and from engine to engine, in the pumping engine business. Then by noting and remembering the action and results under different and various conditions of pressures, speeds, capacities, etc., a line of procedure in any and certain cases which arise, is finally established. After some years of experience, trials, and tribulations, the approximate relation between steam economy and design is fairly well ascertained; the final results seeming to show that purely scientific men saw the right road long ago, but from the lack of opportunity for practical application could not travel far enough and could not live long enough to find out exactly what kind of a machine was available for the purpose in view.

The savants who many years ago understood a good deal

about the strength and quality of materials, and the possibilities of heat, were somewhat in the position of the man who inquired of the gardener at a certain famous educational institution in England, how he produced such a beautiful lawn, and was answered:

"You must first level and prepare the soil very carefully, then get and sow the very best seed to be had, and then mow and trim the lawn a couple of hundred years or more, and you will get the results you are looking for."

The generation of steam in boilers and the use of that same steam in pumping engines, are two totally different operations; independent, and distinct. And from this it follows, that the boilers are not responsible for the steam used and wasted by the engine, and that the engine is not responsible for the cost of the steam produced by the boilers. But in the early days back of the last century, when the pumping of water was a limited industry, before men had looked quite so far into all sides of the question, and had in a hard-headed, practical way, observed only the coal shoveled into the boiler furnaces and the amount of water made to run away from the top of the mine by the pump, took hold apparently of only one end of the problem in real earnest, and that was the engine end of it. Even when the steam pressure was raised for better economy and more work, it was on account of the engine, and the boilers simply had to meet this new condition. Therefore it was natural enough that the basis of 100 lbs. of coal should finally become established as the unit of what was consumed, to be charged against the work done by the engine. In later years, the kind, quality, and condition of the fuel used for generating steam, arising from the wide application of pumping machinery, took the engineer far afield and gave him all the way from natural gas, oil, tan bark, and saw dust, to the very highest qualities of anthracite and bituminous coal, and of course the absurdity of talking about 100 lbs. of coal as a basis for duty tests became apparent.

After a while it became to be generally recognized that 100

lbs. of good Cumberland coal would readily evaporate 1,000 lbs. of water into steam, and so the unit of 100 lbs. for the consumption of fuel was based upon the assumed evaporation in the boilers of 10 lbs. of water per pound of coal; the actual record of coal for a duty test being ascertained by weighing the feed water which was to be made into steam in the boilers, and dividing the amount of water so found by 10, the result being taken as the weight of coal which would be consumed on account of the engine, upon the basis of 10 to 1 evaporation in the boilers. It did not require very much time after going so far to perceive that a still shorter road to a statement of the results obtained, would be to base the duty developed directly upon the unit of 1,000 lbs. weight of steam delivered to the engine. The use of this basis, separating, as it does, the performance of the boilers from that of the engine, is now very much used, although still another basis has been brought forward, and that is the million heat unit basis, which is simply another way of using the old coal basis of 100 lbs. by assuming that good boiler work consists in utilizing 10,000 heat units per pound of coal by the boilers, or 1,000,000 heat units per 100 lbs. of coal. The method of testing a pumping engine upon the basis of work done by the main pumps, per 1,000 lbs. of dry steam supplied to the steam cylinders, jackets, reheaters, etc., appeals to the writer as the simplest and most satisfactory, and also the method most in line with the actual performance of making and using of steam for the pumping of water. The heat unit basis is, perhaps, strictly speaking, the more scientific, and a method which builders of pumping machinery will want to know the results of. But, after all, the steam must be made at some certain cost entirely independent of either method of testing, and the feed water must be weighed in either case so as to know how much either of steam or of heat goes into the cylinders and appurtenances of the engine.

First, taking up the 1,000 lbs. of steam basis, reckoned from the weight of feed water; when the contract requirements for duty are based upon the quantity of steam sent from the

boilers and used in the engine, great care must be taken to accurately determine this amount; all connections between the boilers which supply steam to the engine, and other boilers in the station, must be either broken or closed with blank flanges. The water used as feed water for the boilers supplying steam for the test, may be drawn from the force main or other suitable source, and discharged into weighing tanks or barrels placed upon platform scales, the scales having been tested by some sort of official weights; and from the weighing barrels the water is discharged into a tank or barrel from which the supply for the boilers is taken. Provisions must be made for either obtaining the weight of the drainage from condensed steam within the main steam pipe, or for returning this water directly to the boilers; but the drainage of condensed steam from the steam jackets and receiver coils must be allowed to escape as part of the steam consumed by the engine.

During the test, all steam used for mechanical stokers, and for any other purposes not fairly chargeable against the work done by the main engine, must be taken from a boiler separate from the steam supply for the main engine, and this brings in the question of feed pump supply where the boilers are supplied with water by a pump independent of the main engine. The complete pumping engine must feed its own boilers, and this can be readily and often is done by an attached feed pump on the main engine; and so this amount of work, amounting to about one quarter of one per cent of the total power, is in such case provided by the pumping engine itself. When an independent steam pump is used, there would be a consumption of steam in proportion to the work of feeding the boilers, very much in excess of what would be required by the main engine for the same amount of work with an attached feed pump. This use of steam by an independent pump would unfairly and adversely affect the duty record of a pumping engine, so a compromise is in order; and a reasonable one is to charge the main engine with the work of feeding the boilers by deducting the foot pounds

represented by such an amount of work, from the total work shown by the main engine, before dividing the total foot pounds for the duty results. The writer distinctly remembers such a controversy some years ago when testing a pumping engine in conjunction with Professor David M. Greene. The professor represented the city buying the engine, and the writer represented the builders; and in summing up the articles, so to speak, just before the test was commenced, the absence of an attached feed pump became known. The city's representative and the builders' representative promptly began a lively debate, and at one time it looked as though the third man provided for by the contract in case of dispute would have to be sent for.

The city's man said that the engine was not complete unless it fed its own boilers, and in the absence of its own feeding apparatus the engine must take the consequences. The builder's man said that the duty was to be based upon the steam consumed by the engine itself and not by a steam devouring direct acting steam pump requiring 200 lbs. of steam per horse power hour. Finally the suggestion was made, — modesty forbids recording by whom, — that the foot pounds found by multiplying the weight of the feed water, by the head in feet represented by the boiler steam pressure, be deducted from the total work of the main pumping engine before the final calculation for duty was made, and then the steam for the independent steam pump to be taken from a boiler separate from those supplying the main engine.

After due consideration, and a proper amount of suspicion displayed by the other man, this was agreed upon and then the clouds disappeared. The principle involved in this adjustment is that a pumping engine under this sort of a test should do certain things that fairly belong to its work, and if it is not fitted up to do all of the proper auxiliary work, then have it done in some convenient way, and charge such work actually done to the main engine, but do not handicap the machine by more wasteful methods than it employs itself. The equity

of the matter is that the buyer can get the benefit of economical attached auxiliaries by paying for them, and the original price for the engine would have been higher if the auxiliaries had been attached to the main engine in the first place.

However, this rule would not apply where an independent condensing apparatus is used in connection with a pumping engine, as the driving of the air pump is too much a part of the work of the main engine to be ignored. Too large a share of the economy of the machine comes from the use of a vacuum, to omit the steam consumed by the independent air pump from that charged against the total work done; but of course in the case of the independent air pump if the steam is charged the main engine must be credited with the work represented by the operation of such an air pump, as when the air pump is attached to the main engine the work done by the air pump is included in its own.

It is fair and proper to determine the leakage from the boilers and piping under pressure during the engine test, and deduct the amount found from the steam consumption, as it is plain that water fed to boilers for steam making cannot be fairly charged to the engine if a part of such water gets away before it is received by the engine as steam; and the records show that no boiler and its pipings are absolutely steam tight. The lowest record of leakage of boilers and pipes known to the writer is sixteen hundredths of one per cent; another low record is seventy-four hundredths of one per cent; and there are plenty of boilers and piping easily as high as two per cent and more. In the absence of an actual test for such leakage, it is no more than fair to allow, say, one per cent for boiler and steam pipe leakage, when the duty of the engine is desired accurately and in the absence of prohibition on this point in the contract.

If the boilers are to be measured for loss before the engine test is commenced, as good a way as any is to measure the amount which the water will drop in the glass gauges by drawing from the boilers certain known weights of water, and then note the difference in water level in the glasses corresponding

to such quantities withdrawn. Then bring these readings of water levels to an average basis of one inch and correct if necessary for difference in temperatures; and this will show that for each inch and fraction of an inch of change in water level in each boiler, a certain weight of water would have to be accounted for, either above or below any certain fixed mark, for a range within the ordinary uses of the boilers.

After so fixing the rate of change in level and weight in the boilers to be used for the engine test, a test for leakage should be made for these boilers, by placing them under the working pressure for, say, ten hours, and noting the loss in water level as indicated by the drop of the water in the glass gauges. Any steam condensed within the main steam and connections during this test can be sent through a pipe coil surrounded by cold water and then weighed, the amount so found to be deducted from the amount noted as disappearing from the boilers in the glass gauges.

A test should also be made to ascertain how much steam would be lost from the main steam pipe by the operation of the calorimeter, it being necessary to blow a certain amount of steam from the outlet of the calorimeter to ascertain the percentage of moisture in the steam going into the engine, where under the contract the engine is entitled to the credit of such percentage of water contained in the steam, and also a credit for the amount of steam wasted in operating the calorimeter.

Then, in a 1,000 pound feed water test, the net dry saturated steam used by the engine during the duty test would be ascertained by deducting from the gross weight of feed water sent to the boilers, including what amount may be needed at the close of the test, if any, to make up the correct boiler level in the gauge glasses; the leakage ascertained; calorimeter waste; any other observed leakage before the steam reaches the main throttle valve; and the amount of water represented by the entrainment within the steam passing the engine throttle valve.

Sometimes in competition pumping engine builders will try to outdo each other in the line of guaranteeing high duties,

and then bringing forward the argument that a higher duty at a higher price is a better engine than the reverse proposition; and sometimes the engines are guaranteed so high that it requires all sorts of expert manipulation to meet the contract when the day of testing the engine at length rolls around. In such cases the entrainment of water in the steam is eagerly looked after and very sharply too, for it is a direct credit, and the larger it can be made, the more the steam divisor is reduced in the final calculation, which tells the tale of success or failure in the attempt to make the required duty. The story is told of a representative of one of the large makers of pumping machinery, who must have been a real expert, by the way, who perceived that the results were alarmingly close, and, as he had exhausted every allowance he was entitled to, wiped the perspiration from the faces of the firemen and running engineers, and by sustaining the claim that this should be added to the entrainment because its presence was due to the heat of the boilers, just managed to pull through on the duty requirements.

If the 1,000 lbs. of steam test is upon the basis of dry saturated steam, then any superheat in the steam used must be charged against the engine, by adding to the net weight of dry saturated steam ascertained in the manner already given, the weight of steam that could be made by the heat units shown by the amount of superheat observed. This would be done by multiplying the degrees of superheat by 0.48, which will give the units of heat in one pound weight of the superheated steam over and above the regular amount of heat due to dry saturated steam; and this multiplied by the net weight of steam used, and then divided by the difference between the total heat units in steam at the observed pressure and the heat units in the feed water where it enters the boilers, will give the extra pounds weight of steam to be charged against the engine in addition to that found in the usual way.

When steam shows more moisture than it is entitled to according to the pressure and temperature of dry saturated

steam given in a proper steam table, it is said to contain entrained water; and therefore if the contract duty is based upon dry saturated steam, a correction must be made in favor of the engine, because there is more weight in the steam charged against it than dry saturated steam can show. As the latent heat in the steam is the same percentage short as the percentage of moisture, the correction can be made by multiplying the latent units of heat from the steam table according to the pressure observed, by this percentage expressed in decimals, and the result will show how many heat units must be deducted from the total heat in the steam to give the actual heat units in a pound of the moist steam. Then this result divided by the total heat given in the table will show what percentage of a pound of dry saturated steam the actual amount of heat will make. This latter result subtracted from one (1) will indicate the percentage to be deducted from the net weight of steam obtained during the test already referred to.

Second, taking up the million heat units basis, it will perhaps be well to remark that a heat unit, or a unit of heat, commonly known as a British Thermal Unit, is the amount of heat which will raise the temperature of one pound of fresh water one degree, on the scale of the Fahrenheit thermometer, that is the ordinary thermometer we are used to seeing every day (water at a temperature of 39 degrees).

When the requirement for duty is calculated upon the heat unit basis, it is made to cover all of the heat used by the main engine and its appurtenances and auxiliaries; including the steam cylinders of the main engine, steam jackets, reheating coils, steam feed pump if such is used, the independent air pump if one is used and driven by steam, and every other appurtenance and appliance or apparatus using steam, and necessary to the operation of the pumping engine. In the steam supply for the engine, allowance must be made for moisture or for superheat as the case may be, as the heat unit basis demands that the real total heat of the steam be determined.

The heat units consumed by the engine is the difference

between the heat units sent into the boiler in the feed water, and the heat units taken out of the boiler in the steam which the engine uses. This difference is supplied by the burning fuel, and it is the total amount of heat given up by the fuel for steam, regardless of the quality of the fuel itself, and which is divided into blocks of 1,000,000 heat units each. From this the work done by the engine per million heat units is found by dividing the total work done by the engine in foot pounds, by the total number of heat units shown to have been supplied which will give the foot pounds of duty for each unit, and this number multiplied by 1,000,000 reduces the result to the basis desired, which is the number of foot pounds of work done by the engine for each 1,000,000 British thermal units consumed by the engine and its appurtenances.

If any heat is obtained from the boiler smoke flue or connection by the use of a coil or similar device connected with any part of the engine, so as to give the engine the advantage of heat escaping from the boilers, this heat must be added to that which is indicated by the steam consumed; because it will be heat from the combustion of the fuel just the same as though it was received by the engine in the form of steam; also, heat obtained and absorbed by the feed water from coils or heaters in the boiler flues and connections is to be treated as an addition to that obtained over and above the heat put into the feed water by the engine itself and its appurtenances. In fact, no allowances are to be deducted from the heat charged against the engine, which comes from any source excepting the engine and its appurtenances.

As the heat unit basis depends upon the actual amount of heat in the steam used, the total heat of the steam found in a steam table is corrected for entrained water, or moisture, and also for superheat. Steam contains moisture because there is not enough heat present in the steam to make it dry; and as the sensible temperature, or the temperature shown by the thermometer, is always in evidence according to the pressure, it follows that the shortage is in the latent heat, or

that portion of the heat in the steam which the thermometer does not show, but which is absorbed at the time of evaporation in the boilers. Therefore, if there is one per cent of moisture or entrained water shown to exist in the steam, the latent heat is one per cent short; so the units of heat given as latent in the steam table are to be discounted one per cent, and the total heat in the steam is to be taken at one per cent of the latent heat short of the total amount given in the steam table, per pound weight.

Another way for determining the amount of heat consumed by the engine and its appurtenances, when there is moisture in the steam, which will give the same result as that already given, and therefore furnish a check on the first method, is as follows:

After the correct weight of the feed water has been found, subtract from this weight the percentage of moisture in pounds, shown to exist in the steam; the weight of the moisture or entrained water is found by multiplying the correct weight of the feed water by the percentage of moisture expressed in decimals. The remainder given by this subtraction will be the weight of dry saturated steam possible to make with the heat present.

Next multiply this weight of dry saturated steam by the total heat units per pound found in the steam table; and then multiply the weight shown by the percentage of moisture, by the units of heat per pound of water at the temperature due to the pressure observed. The sum of these will be the total heat actually supplied to the engine, which must be corrected for duty calculations by subtracting the units of heat sent into the boilers by the various supplies of feed water.

If the steam is superheated, it indicates that whatever number of degrees the steam is shown to be hotter than the temperature called for by the steam table due to the observed pressure, there is present in the steam 0.48 of a unit of heat for each degree superheat above the normal temperature of steam called for by the table. The heat to be added to the

account against the engine by reason of superheat, is found by multiplying the number of degrees superheat by 0.48 and then multiplying this product by the net weight of the feed water shown to have been used for steam.

When the feed water is combined from all sources about the engine and sent to the boilers as one body, at one average temperature, this temperature is to be taken at some convenient place as near to the boilers as practicable. But when there are several separate temperatures of feed water sent by different pipes so as not to conveniently combine together, as, for example, one from the hot well, one from the jacket condensation, and from reheater coils, or other sources at different temperatures, they are to be treated separately on account of the heat units due to the various temperatures to be compared with the steam going to the engine.

Then the total number of heat units, or British thermal units, consumed by the engine and its appurtenances is taken from the weight of the water sent into the boilers by the main feed pump; the weight of the water sent in from the steam jacket drainage; the weight of the water sent in from the reheater coil drainage; and the weight of any other water sent to the boilers at a different temperature from any of the other supplies. These different weights are to be multiplied by the total heat in the steam, calculated from the different temperatures of the different sources of feed water; and the result obtained in heat units added together for the total heat units consumed.

Even under the heat unit method for calculating the duty of a pumping engine, the contractor sometimes guarantees a rather high duty, which of course has to be shown somehow or other by the expert, or at least earnest endeavors to show that everything is all right have to be put forth now and again. Another story is told of an incident which happened in connection with one of these heat unit duty tests, also involving one of the larger manufacturers; in fact, the large builders are the ones who have such bright and attentive representatives.

It seems that this test was about half finished and it had become apparent that the guarantee was a very high one for this particular engine, the chances evidently being against the machine. There was a cistern outside of the boiler room, and one of the firemen just going off watch accidentally fell into it, but suffered no damage beyond the thorough soaking of his clothes. The weather being rather raw he feared taking cold, so returned to the boiler room and backed up near to the hot front of one of the boilers making steam for the test; but just at that time the expert came into the boiler room to look after the water level, and taking in the situation at a glance, ordered the wet fireman away from the boiler front, telling him in no stinted terms that this test was a heat unit test, and there were no heat units to be spared for drying clothes or for any purpose whatever outside of running the engine.

Having provided for finding out the steam or heat consumption during the duty test, the plunger load, or the work done by the pumping engine, comes in for equally careful attention. The measuring of the work done carries with it the determination of the quantity of water delivered into the force main of the pump, and although several and various ways have been advocated and tried from time to time, including weir measurements, Venturi meter tubes, nozzles of known delivery under certain conditions, etc., it is becoming pretty generally conceded by disinterested people looking for the facts, that in a properly designed and constructed displacement pump, the plunger displacement will give closer results in a matter where perfection is impossible, than any other method. In fact, aside from the slight leakage and loss inherent in the very best of pumps, the pump itself is a perfect meter; and there are loss and error in all means for determining quantities of water outside of the pump. There is another advantage in the use of the pump itself for measuring, and that is, that the error, whatever it may be, is always on the same side of the calculation, whereas in about everything else used for measuring the error may be on either side of the account, especially so

with weirs, and a weir would make itself ridiculous and every one connected with it as well, if it indicated more water by having too high a velocity of approach than the plungers could displace. Also, experiments tried by the writer have at different times demonstrated that the record of a measuring instrument placed in connection with the suction flow will vary at all times, with a record of the same instrument placed in connection with the discharge flow, and vary greatly sometimes. The slight leakage around outside packed plungers can be measured down to a very fine point and the correction made for readings before and after the water has been pumped. Allowing, then, such corrections, the quantity flowing through the suction ought to be the same as in the delivery if the quantity is really and accurately measured, or at least correct within the limits of trained observation.

Another experience is also recalled, where there was an opportunity of comparing the delivery of a large and perfectly constructed weir according to the Francis formula, and two Venturi meters. Very great precautions were taken in making the comparisons, and considering the amount of care and precision given to the work, there is no reason to doubt that every possible error in manipulation and observation was avoided. There was a disagreement of about 2 per cent between these two methods of measuring water, and this disagreement is more than any well regulated pumping engine will show under good conditions, with as clear water as in the above mentioned test, and with outside packed plungers. No reflection is intended upon either of the above methods or against any other method; the desire is only to get at the facts as closely as possible; and where weirs or meters can be used in conjunction with plunger displacement, it is by all means the thing to do in endeavoring to arrive at the facts.

The work actually done by the steam engine portion of the machine where the plungers are moved against a water pressure or load, is just the same as though none of the water escaped going into the force main. The water that escapes past the

plungers or through the valves, goes out under the working pressure, and the power developing end of the engine knows no difference as to the means and way of escape in quantities up to the ordinary losses in such machinery. It is not difficult to determine very closely the amount of loss or leakage, if in the interest of the buyer this is desired; but as such losses in a well made pump in good order are very small, there is really very little use in spending money beyond what is demanded by proper surroundings for the engine. If a pump is well designed and made, with the size, number, and lift of its water valves in good proportion to the capacity, pressure, and speed, for clear fresh water, the builder should not be held responsible as to the amount of water it will actually pump under bad conditions; and it will be to the interest of the buyer to have his conditions brought as nearly to the clear water basis as can possibly be done. The pump is not responsible for chips, rubbish, fish, and other things not meant for a pump to handle, and the buyer and user is greatly interested in seeing that the pump well is properly screened and kept clear of all substances not intended to be in there. So, according to the writer's views, the most practicable way to meet this question is to calculate the plunger capacity 5 per cent above the normal contract capacity in order that the buyer will be very certain to get the full amount of water figured upon for actual use, or waste, and then make the duty test squarely upon the plunger displacement. If all makers bid upon this basis under competent specifications, the buyer will get just about what he is looking for, so far as capacity is concerned; the duty shown upon test by plunger displacement will be a good conventional guide for future operations, and with a proper displacement pump, will under good conditions be as close to the facts as any other practical method will carry him.

In finding the plunger load, a mercury column is decidedly the best and most accurate way, and it is worth while taking some trouble to fit one up for a test. Tested water pressure gauges, compared with some permanently constructed mer-

cury column, where one is available; or with some form of dead weight apparatus, as, for example, the Crosby dead weight testing machine, will answer very well for reading the load, if it be impracticable to set up a temporary mercury column in connection with the engine being tested. In most pumping stations there is height enough from the basement floor to a moderate distance above the main engine room floor, to place a mercury column which can be well made of iron pipe most of the way, and with several feet of glass tube near the upper end; but the mercury should not come in contact with brass. A very perfect reading scale can be easily attached to a board near where the mercury level would naturally come, when the tube is filled up with the mercury to the working load.

A bench mark can be established by means of a surveying instrument, and to this bench mark all vertical measurements can be referred. The elevation of the water in the pump well can be read from a graduated board set to correspond with the established bench mark, in such a manner that the reading of the mercury column or pressure gauge, whichever is used, can be added to the reading of the well gauge, and so give the total head in feet against which the plungers operate. But there are generally certain corrections for temperature, disturbance of water levels, pressures, etc., etc., to be made, for finding the actual working head under the conditions found at different pumping stations.

The diameter and stroke of the plungers submitted in a bid, and shown in the plans for a pumping engine, should be verified before positive statements about a duty test are made. These measurements should be made by instruments of precision, such as micrometers, steel scales and tape lines, from well recognized makers. The actual dimensions of the work as furnished in the engine is generally found to vary somewhat from specifications, even when coming from the best concerns, although, to the credit of the builders, generally varying in going a little over the full measure; as, for example, in two cases known to the writer, of recent occurrence, one a

triple and one a cross compound pumping engine. The plungers of the triple machine were high pressure 28.253, and intermediate pressure 28.253, and low pressure, 28.259 inches diameter; the size called for in the contract was 28.25 inches, for all. The stroke for high pressure 60.05, and intermediate pressure 60.01, and low pressure 60.03 inches respectively; the stroke called for in the contract was 60 inches. In the cross compound, the plungers were, high pressure 18.263 and low pressure 18.279 inches diameter, where 18.25 inches diameter was called for in the contract for both plungers; the stroke of both plungers was 48.125 inches where 48 inches was called for in the contract. These differences are very fine, but they show that these engines would meet the displacement capacity without any doubt whatever.

The capacity of the plungers per revolution of the engine, in cubic feet, having been established from the diameter and stroke found by the measurements, the weight of a cubic foot of water must be decided upon, and from this the weight of water per revolution is readily obtained. The number of U. S. gallons per revolution is obtained by multiplying the cubic feet by 7.4805, the number of U. S. gallons of 231 cubic inches each, contained in one cubic foot of 1,728 cubic inches. In ascertaining the weight of a cubic foot of water in the pump well, the temperature must be known, and the weight corresponding to the observed temperature may just as well be taken from some well recognized table of weights of water at different temperatures, as it is impossible to get any vessel measured closely enough for a cubic foot or a cubic inch, or any other capacity, for any two, or any other number of men to find exactly the same. Ten of the world's greatest authorities all differ slightly in the weight of a cubic foot of water at the same temperature. They differ very little, to be sure, and very likely as much from the difficulty of getting the standard vessel of measurement the same in all cases, as from anything else. It is impossible for one man to make two vessels alike in size, and also for two men to make one each and have both

the same size. Such duplication is beyond the powers of human faculties; but they get them near enough for practical purposes, and very much nearer than the results can be read and recorded in a duty test of a pumping engine.

In an ordinary duty test a slight disagreement does not matter so very much, as to the weight of a cubic foot of water, so long as the contract duty is fully met; but in a duty test where money as a bonus or a penalty is to be charged against or credited to the contractor, according to the duty obtained by the engine, some absolute figure must be agreed upon for the weight, because only one result is permissible. The writer after many observations, and consulting many authorities upon the subject, has arrived at 62.42 lbs. for the weight of a cubic foot of water at or below 48 Fahrenheit, and above that temperature is governed by circumstances bearing upon the case at the time of a test.

From the weight and capacity per revolution the result per minute, hour, day, or any other period of time, may be easily computed; and the weight of water so found multiplied by the correct total head against which the plungers have worked during the duty test, will give the foot pounds of work done, and against which the weight of steam, or the number of heat units, a e to be charged in finding the final results.

Before the test is commenced, the various substances and instruments needed in the work should be carefully considered and their respective values agreed upon, so that a unanimous decision can be reached when the test is finished. And it is also necessary to fix certain points of elevation so as to have proper data upon which to calculate at any time needed; such points of elevation should be observed and recorded.

Sometimes in testing a pumping engine for duty, instead of weighing the ingoing feed water, the record of steam consumption is obtained by weighing the outcoming water of condensation where a surface condenser is used with the engine. In such a case the steam consumed by the engine and its appurtenances is taken as the weight of the condensed steam delivered

by the air pump from the surface condenser; the weight of the condensed steam rejected by the reheating coils of the receivers between the cylinders; and the weight of the condensed steam rejected by the steam jackets of the engine, plus any amount observed as leakage or loss before the condensed steam reaches the weighing tanks or barrels, and plus the amount that may be used by any independent apparatus necessary for the operation of the engine itself. All of these amounts so found are to be treated the same as similar quantities mentioned in the feed water method for determining the steam consumption, or for determining the heat supplied to the engine and its appurtenances. With this method of finding the steam consumption by means of the condensed steam, there are used the same means for determining the plunger capacity and load, as in the methods already described.

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